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ENGINEERING REPORT

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1.0 INTRODUCTION

THIS
The objectives of Contract ~~DAK10-79-C-0096~~ were to design prototype auto-loader and recoil systems for the 155mm howitzer mounted in the M109A2 vehicle. These systems were to require the minimum changes possible to the existing vehicle and were not to impair the 360-degree traverse and 0- to 75-degree elevation capability of the weapon. Projectiles and powder charges would be stored and fed automatically by either powered or manual operation of all functions, with a burst rate of three rounds in 10 seconds and a sustained rate of four rounds per minute. The howitzer would have a sliding breech and a fixed or variable recoil length. The primers would be fed and ejected automatically as well. The system should be recoil operated with instantaneous selection of the various powder charges and projectiles from the gunner's station. The ammunition stowage racks should hold more than 43 rounds and permit setting of the fuzes in the stowed position.

The recoil system should be of modular design with each module replaceable without draining oil or gas. All cylinders should be designed for a maintenance-free life of 10,000 rounds. Each pair should be identical, and the system should function on one each of the recoil and counterrecoil cylinders. Both systems should maintain simplicity and adequate reliability while operating over a temperature range of -50°F to $+160^{\circ}\text{F}$.

The following is a report of the methodology and design of the autoloader and recoil system designed by Pacific Car and Foundry Company (PCF) to best fulfill the above requirements in the most practical and reliable manner.

2.0 AUTOLOADER

2.1 RATIONALE

The original proposal concept provided a stowage rack mounted across the rear of the vehicle with the projectiles and powder raised from the stowed position and transported along their respective sides to a feeder which held multiple rounds. The magazine type feeder on each side of the gun would move up to the stowage rack to load and then down to the gun position to present the projectile and powder to the rammer. During the initial evaluation of this design, questions arose as to its ability to perform the necessary functions within the firing cycle constraints. Initially, the idea of a multi-round magazine that would follow the weapon in elevation for the burst-fire rate seemed to be the best method of reserving the maximum allowable time for loading a projectile and powder charge. However, several compromises were necessary with this system. A prefilled magazine of heavy projectiles and charges of one type presented the necessity of removal by hand should the fire order be changed or cancelled. The feed system required projectiles and charges to be lifted from the stowage racks to cab roof level and then moved forward to the magazines, which would lower them to the weapon elevation. Also, the system required all projectiles to be stored on one side of the vehicle's longitudinal center, creating a gross imbalance. Depending on the amount of ammunition used, this imbalance would change greatly as rounds were consumed, requiring a sophistication of the vehicle's suspension system.

Certain design parameters become apparent during this initial study of the autoloading system. To conserve energy and reduce time, it would be desirable to move the projectiles and charges as short a distance as possible

while attempting to use gravity to the best advantage. A gravity feed system was considered. However, a study of such a system indicated a reduced reliability. A system which would advance one type of projectile and charge at a time would eliminate the necessity to remove unused rounds. The stowage racks should retain the projectiles and powder in a positive manner during recoil and vehicle operation. Also, the system should stow the projectiles as low and evenly across the vehicle as possible to maintain an even trim and low center of gravity for the vehicle. At least five types of projectiles and charges should be readily available. Also, a modular design of the stowage racks would enhance reloading capabilities as well as provide greater reliability.

The resulting design presented in this report reflects a departure from the proposal for a more effective and reliable system, while meeting the objectives of the contract as well.

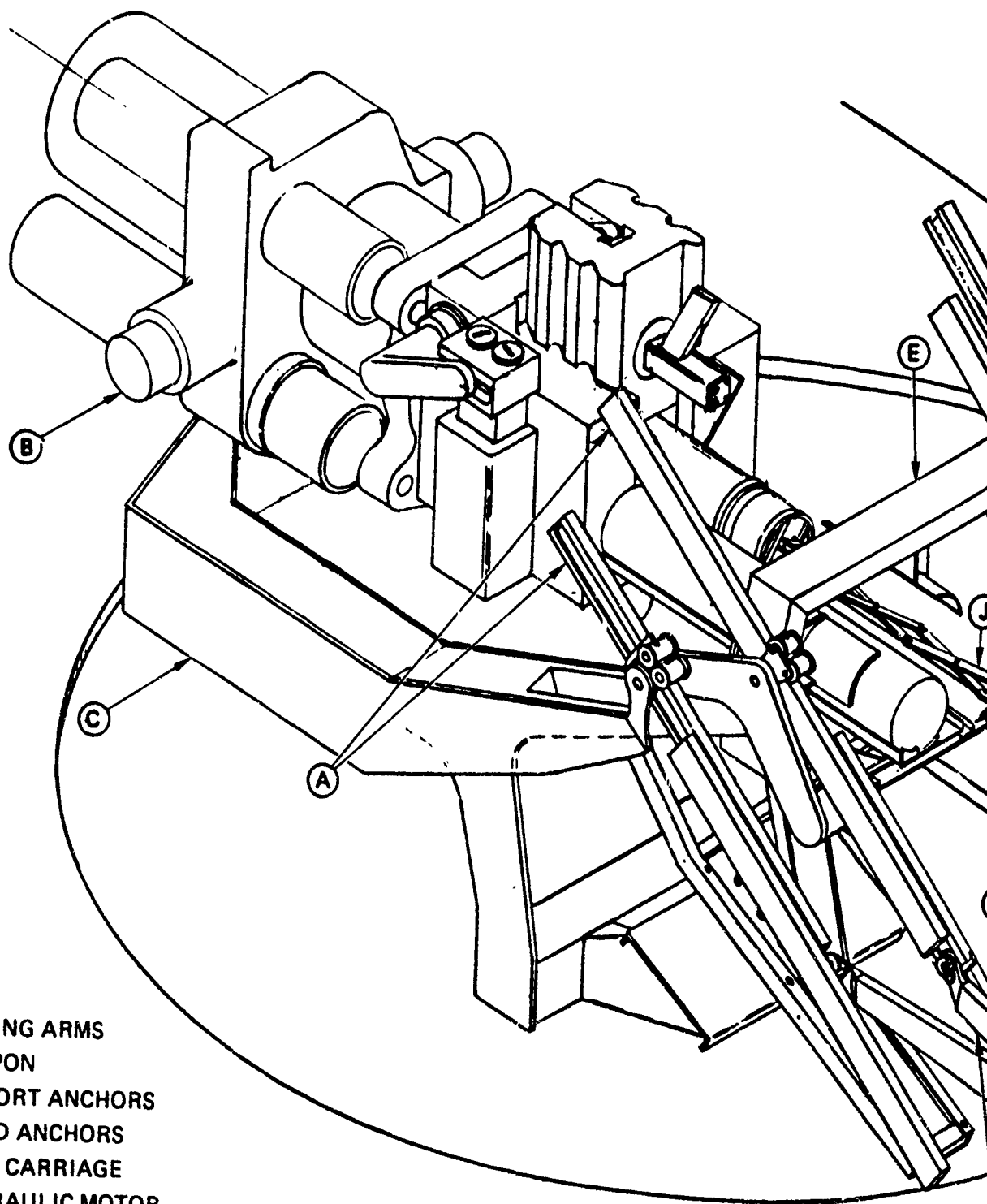
2.2 DESIGN

2.2.1 Autoloader

The autoloader design evolved from the combination of requirements and trade-offs necessary to produce a system that would meet the required firing rate and yet be operable by hand. The preliminary conceptual design assumed that the firing rate could be met only by using a magazine type loader holding two or three projectiles and powder charges and moving in elevation with the weapon. During the initial design phase of this system, it became apparent that although the system would meet the requirements, it would be desirable to eliminate certain features if possible. The necessity to elevate the ammunition from the feed racks to the reload point for the magazines would pose a

problem for manual operation. Also, the mechanism would be fairly complicated, and a weapon stoppage for any reason would require removal of the ammunition in the magazines by hand. Also, multiple handling of the ammunition reduces the system's reliability. In addition to maintaining the proper vehicle trim, the projectiles should be stored completely across the vehicle and as low as possible. To accomplish this, it would have been necessary to reduce the types of rounds carried and complicate the feed system considerably; therefore, concepts for alternate methods were evaluated. A system that would advance one round at a time would be desirable. However, to do this would require a system that would follow the gun during full elevation and yet accept ammunition from a fixed point.

In the design resulting from this study, a simple system of sliding arms (see Figure 2-1, Item A) was devised so that during elevation of the weapon (see Figure 2-1, Item B) about the trunnions, the support anchors (see Figure 2-1, Item C) which are firmly attached to the gun, provide a fixed point for the telescoping arms at all elevations. A fixed reference point at the vehicle is provided by anchors (see Figure 2-1, Item D) attached to the turret. The geometry is such that as the gun changes elevation, the fixed anchors at the gun and the feeder lengthen or reduce the length of the arms (Item A) appropriately. These arms provide a track for the feed carriage (see Figure 2-1, Item E) to travel on. Since the carriage is firmly affixed to the tracks and due to the geometry of the followers on the carriage, the carriage will always be level at the feeder position and assume the angle of the gun when moved to the gun position. The carriage is powered by a hydraulic motor (see Figure 2-1, Item F) which drives a chain (designed to become rigid under compression) on each side of the carriage (see Figure 2-1, Item G). During retraction the chains will bend in one direction



- (A) SLIDING ARMS
- (B) WEAPON
- (C) SUPPORT ANCHORS
- (D) FIXED ANCHORS
- (E) FEED CARRIAGE
- (F) HYDRAULIC MOTOR
- (G) CHAIN
- (H) RAMMER MODULE
- (J) RAMMER
- (K) MODULAR AMMUNITION RACK
- (L) TRAYS

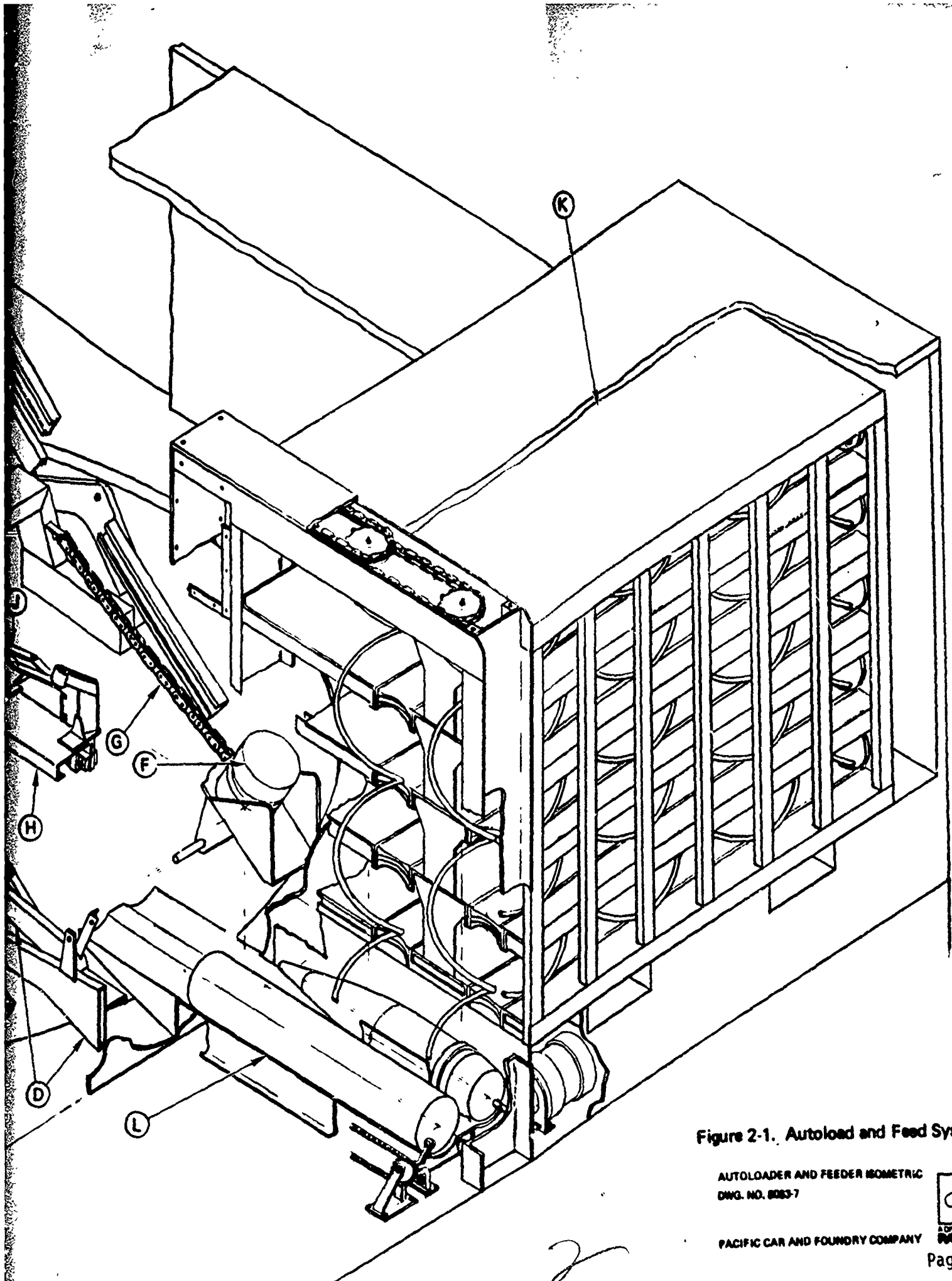


Figure 2-1. Autoload and Feed System

AUTOLOADER AND FEEDER ISOMETRIC
DWG. NO. 8083-7

PACIFIC CAR AND FOUNDRY COMPANY



only, which allows them to turn around the sprocket and store in a small place. Upon arrival of the carriage at the gun (shown level in Figure 2-1 for clarity), the rammer module (see Figure 2-1, Item H) rises from its stowed position below the recoil path of the gun. The rammer module is carried with the gun on a solid structure (see Figure 2-1, Item C) to maintain its relationship with the bore centerline. On its final bit of travel, it picks up the carriage that the powder and projectile have been transferred on and makes the final alignment with the bore for ramming. The projectile is always carried on the centerline of the weapon; therefore, the system is always moving the heaviest items the shortest distance. The carriage moves downward when carrying a projectile and powder, except when loading at gun tube angles of 0 to -10 degrees of elevation. A handcrank at the motor would allow rapid operation of the system in the manual mode.

The rammer (see Figure 2-1, Item J) is of unique design to allow stowage in the least possible space and ram the projectile and powder separately in the least possible time. The rammer is a "flick" rammer in that it accelerates the projectile to above 10 ft/sec to provide proper seating force. After acceleration of the projectile, the rammer leaves the base of the projectile 1 inch inside the chamber. The rammer is capable of velocities above 10 ft/sec if found necessary during testing. Also, the alignment of the projectile and rammer to the centerline of the weapon can be adjusted to eliminate balloting during the projectile free travel. After ramming the projectile, the rammer returns to the retracted position, and the powder charge is moved over in line with the weapon and pushed into the chamber. The rammer is throttled to leave the powder the required 1 inch inside the breech surface. The rammer is retracted and lowered while the carriage returns to the feed position to allow full recoil at any elevation. The breech

will close automatically, feeding a primer and arming the weapon, ready to fire. The weapon is oriented so that the breech block rises when opened and the rammer assembly provides a tray to the rear of the chamber for the projectile and powder charge to travel on during ramming. The rammer cylinder is controlled by means of a valve to provide the speed necessary for projectile seating and reduced force to place the powder in the proper position.

For manual operation or when using the Copperhead projectile, the weapon should be loaded level or at a low angle to allow use of the hand rammer. However, the hydraulic rammer could be used with manual control, if desired, by the addition of an accumulator and hand pump.

A study to determine the feasibility of using the recoil stroke to charge accumulators to self-power the system resulted in the conclusion that such a system would reduce reliability and result in undesirable complexity. Past experience in the recovery of recoil energy has proved unprofitable due to the short time and stroke of recoil (approximately 20 milliseconds and 22 inches of travel). If required, a system to recover energy could be developed in the future.

Although the autoloader is primarily designed to accommodate a sliding breech and modular charge, the system could be modified to use the present rotating breech. The addition of an extendable tray on the rammer assembly to bridge the threaded portion of the chamber would accomplish this. Bagged powder could be used with this system with the addition of a plastic disc to retain the bag in the chamber at elevation.

The autoloader as presently designed will accommodate the 155mm howitzer mounted in the M109A2 vehicle with no major modifications needed for operation up to 45 to 50 degrees of elevation. To achieve a full 75 degrees of elevation, the weapon trunnions would need to be raised approximately 6 inches or the rammer

assembly modified to stow below and slightly to the side of weapon. This would result in a slightly slower rate of fire. Computer time and energy studies of the autoloader mechanism show the ability to achieve the 5 second burst rate within reasonable velocities and energy levels. The computer studies (see Figure 2-2) were conducted using 50 percent of the time to accelerate the mechanisms and 50 percent for deceleration. Additional studies using 60 percent acceleration and 40 percent deceleration and 70 percent acceleration and 30 percent deceleration show that the accelerations and energy levels could be optimized if necessary.

Stress and friction calculations and details of the autoloader are contained in Appendix A.

The design of the autoloader uses fabricated and commercially available parts, rather than castings, extrusions, and special parts, to facilitate the building of a prototype prior to production engineering.

2.2.2 Stowage and Feeder

The stowage and feed system was presented a challenge in the need to provide six different types of projectiles and powder charges, readily available and in any order. The feed system must offer safe reliable stowage for the components, yet provide ready access for changing fuses or charges, and for replenishment. A unique system using a lead screw type device to advance the units in rows was conceived (see Figure 2-1, Item K). Each row would use two lead screws to capture the units in the rack. These same lead screws would advance the units simply by rotating one revolution. To reverse the feed, the drive to the lead screws is reversed. The projectiles and charges would be advanced to an elevator incorporating the same type of lead screws. The elevator would lower the projectile

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45°					
CARRIAGE RUN PROPORTIONS ALLOTTED ARE- ACCEL. = .5 DECEL. = .5 DO YOU WISH TO CHANGE PROPORTIONS OF ACCEL.-DECEL. DIST? YES (1), NO (2).					
TIME (SEC)	CH. STROKE	DIST. (IN)	CHAIN LBN. (IN)	// FORCE (LB)	CHAIN LB. (LB)
ACCELERATION STROKE					
10.6858	7.99934	7.38802	-87.762	-105.428	
11.774	8.88876	-87.762	-87.762	-105.428	
12.862	9.77748	-87.762	-87.762	-105.428	
13.950	10.666	-87.762	-87.762	-105.428	
14.038	11.554	-87.762	-87.762	-105.428	
DECELERATION STROKE					
20.6858	20.7985	24.5365	-183.552	-183.552	
21.774	21.687	24.372	-183.552	-183.552	
22.862	22.576	24.208	-183.552	-183.552	
23.950	23.464	24.044	-183.552	-183.552	
24.038	24.353	23.880	-183.552	-183.552	
CARRIAGE RUN PROPORTIONS ALLOTTED ARE- ACCEL. = .5 DECEL. = .5 DO YOU WISH TO CHANGE PROPORTIONS OF ACCEL.-DECEL. DIST? YES (1), NO (2).					
60°					
TIME (SEC)	CH. STROKE	DIST. (IN)	CHAIN LBN. (IN)	// FORCE (LB)	CHAIN LB. (LB)
ACCELERATION STROKE					
10.6858	1.07747	7.51877	-75.6971	-90.7995	
11.774	2.15494	-75.6971	-75.6971	-90.7995	
12.862	3.23241	-75.6971	-75.6971	-90.7995	
13.950	4.30988	-75.6971	-75.6971	-90.7995	
14.038	5.38735	-75.6971	-75.6971	-90.7995	
DECELERATION STROKE					
20.6858	20.013	11.3033	-204.71	-204.71	
21.774	20.013	11.3033	-204.71	-204.71	
22.862	20.013	11.3033	-204.71	-204.71	
23.950	20.013	11.3033	-204.71	-204.71	
24.038	20.013	11.3033	-204.71	-204.71	
CARRIAGE RUN PROPORTIONS ALLOTTED ARE- ACCEL. = .5 DECEL. = .5 DO YOU WISH TO CHANGE PROPORTIONS OF ACCEL.-DECEL. DIST? YES (1), NO (2).					
75°					
TIME (SEC)	CH. STROKE	DIST. (IN)	CHAIN LBN. (IN)	// FORCE (LB)	CHAIN LB. (LB)
ACCELERATION STROKE					
10.6858	1.35111	7.71004	-50.038	-50.038	
11.774	2.70222	10.4441	-50.038	-50.038	
12.862	4.05333	16.6076	-50.038	-50.038	
13.950	5.40444	22.7711	-50.038	-50.038	
14.038	6.75555	28.9346	-50.038	-50.038	
DECELERATION STROKE					
20.6858	30.1289	30.5387	-211.829	-211.829	
21.774	30.1289	41.5633	-211.829	-211.829	
22.862	30.1289	46.6791	-211.829	-211.829	
23.950	30.1289	51.7948	-211.829	-211.829	
24.038	30.1289	56.9106	-211.829	-211.829	
CARRIAGE RUN PROPORTIONS ALLOTTED ARE- ACCEL. = .5 DECEL. = .5 DO YOU WISH TO CHANGE PROPORTIONS OF ACCEL.-DECEL. DIST? YES (1), NO (2).					
120.099					
TIME (SEC)	CH. STROKE	DIST. (IN)	CHAIN LBN. (IN)	// FORCE (LB)	CHAIN LB. (LB)
ACCELERATION STROKE					
10.6858	18.0148	55.3956	53.9531	-211.829	-211.829
11.774	0	67.5556	61.4314	-211.829	-211.829
DECELERATION STROKE					
20.6858	30.1289	30.5387	-211.829	-211.829	
21.774	30.1289	41.5633	-211.829	-211.829	
22.862	30.1289	46.6791	-211.829	-211.829	
23.950	30.1289	51.7948	-211.829	-211.829	
24.038	30.1289	56.9106	-211.829	-211.829	

Figure 2-2. Autoloader Force and Time Calculations (Sheet 3 of 3)

and powder charge to trays at the lowest level of the system. These trays are in line with the feed carriage in the stowed position (see Figure 2-1, Item L). A set of pushers moves the ammo components forward to the feed carriage upon demand. The simplicity of this system is very desirable (see Appendix C, Drawing C-1, Sheet 4). Also, the system is never required to lift the projectiles and powder charges. They are rolled with a minimum of friction, horizontally to the elevator, where they are lowered to the ready position. The drive motors for the stowage racks are fitted with a handcrank receptacle for manual operation. The elevator motor, which is also used as a brake, is fitted with a receptacle for manual operation, as is the motor that advances the units to the feed carriage. Hydraulic motors are used for compatibility with the turret drive system. Energy and friction requirements are minimized by the use of plastic or nylon coatings on the wearing surfaces of the system. Projectiles and powder charges are firmly held in place at all times, and it is expected that advancement and lowering of the units could be accomplished while the vehicle is in motion or during recoil. However, the system is designed to advance the next ready round during the autoloader cycle.

The stowage and feed concept was so unusual that it was decided to fabricate a half-size model to prove the theory prior to finalizing the design (see Figures 2-3 and 2-4). The model, although unrefined, shows the ability to use the concept and is submitted as part of this report. The racks as designed can be accommodated aboard the M109A2 with a minimum of modifications. It will require a slightly higher bustle and the addition of a set of doors across the back. The doors are designed to lower to a horizontal position to provide a platform for individual loading of projectiles and powder charges. The doors can also be dropped to a vertical position to provide access if a modular rack

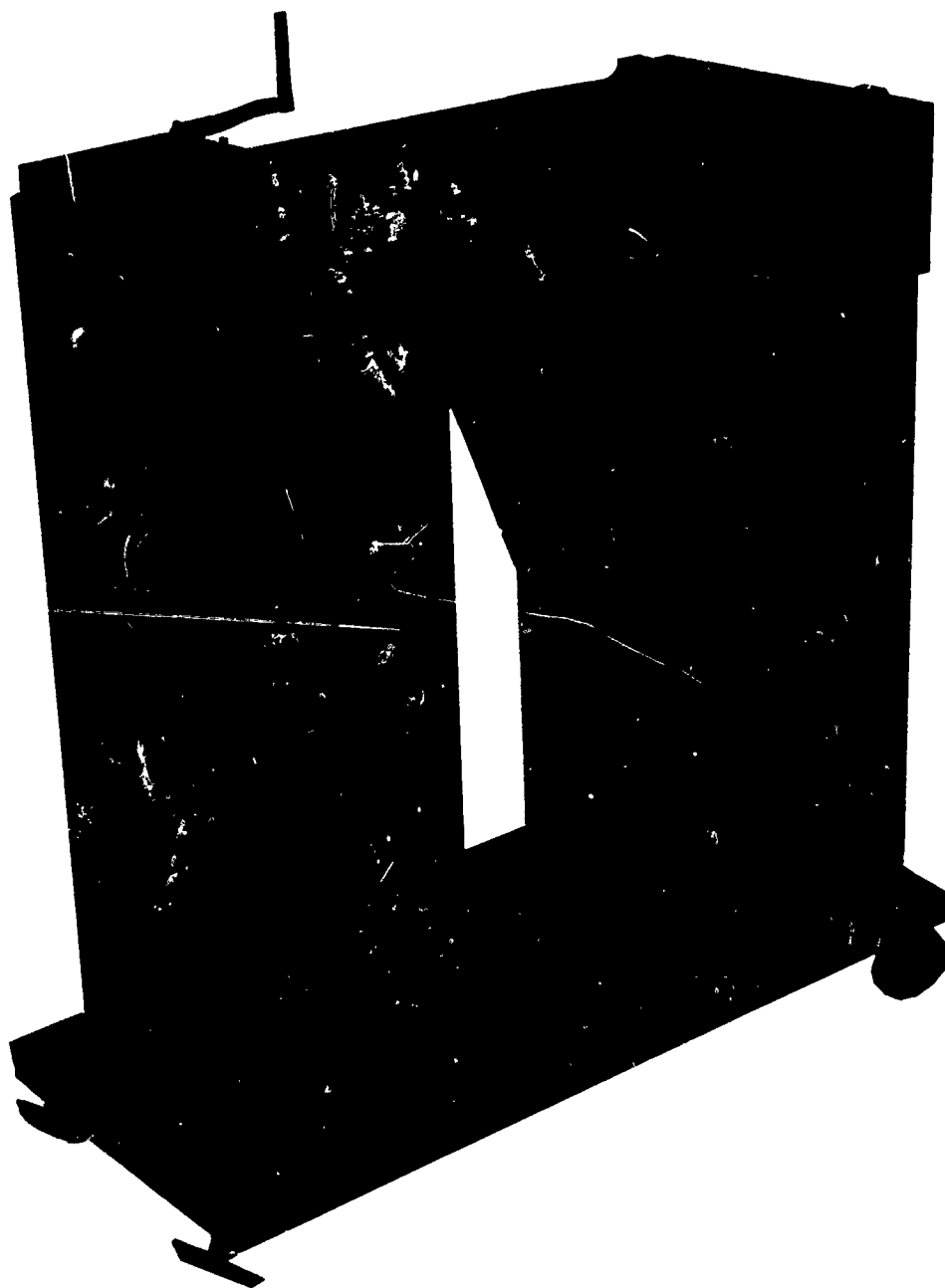


Figure 2-3. Stowage Rack Model

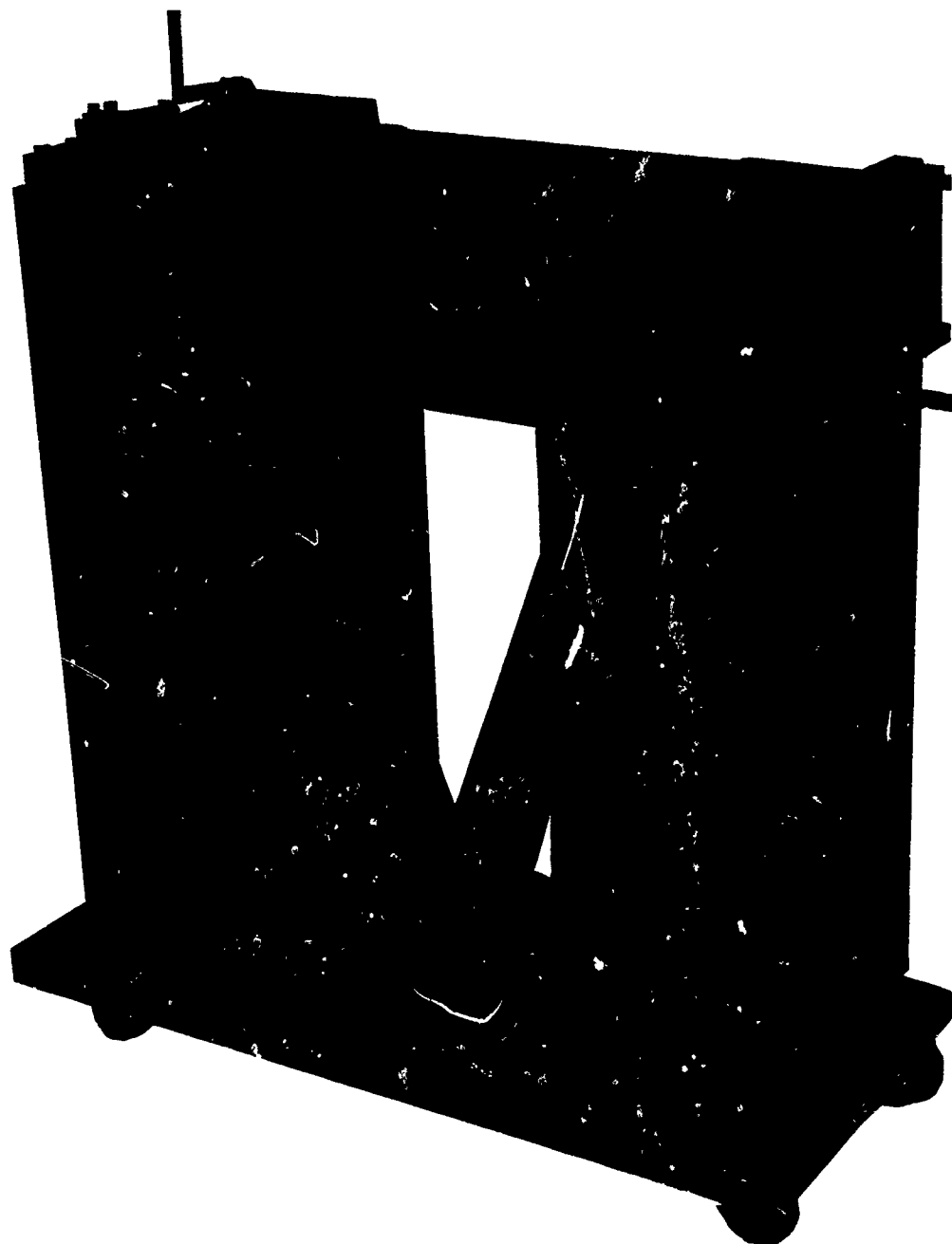


Figure 2-4. Stowage Rack Model

is used. The present configuration will hold 30 complete ready rounds (6 rounds of 5 types). Adaptability to the present M109 vehicle and modular design were the prime considerations for this configuration. A modular rack is shown in Figure 2-1 (K). The elevator section remains with the vehicle as do the drives for the stowage rack. The concept shows fork lift pockets to facilitate removal of the entire rack except the drives. The racks and drives are simple and would be fairly light. The racks would be filled at the depot area and transported to the vehicle directly. Either a fork lift or boom crane could be used to remove and replace the racks. Excess room was left at the bottom of these racks for a test situation; however, 35 ready rounds could be carried by utilization of this area with very little additional modification. Stress calculations and specific details are contained in Appendix A. The system is designed to be fabricated from available material welded, etc., as a prototype. Production design would include the use of castings, extrusions, etc.

2.2.3 Primer Loader

The automatic primer feed system for the 155mm autoloader performs basically the same function as a small arms automatic weapon. However, the shape of the M82 or M119 primer does not lend itself well to autoloading as the sharp square front and the rim at the base create difficulties not normally encountered with regular cartridges. Also, the necessity for the primer loader to operate in conjunction with the breech presents a unique cycling problem. The sliding breech was selected as the prime candidate for the autoloader system. Therefore, the primary emphasis of the design is the use of this type of breech. Although either of the concepts could work with the rotating breech, a rotating bolt type mechanism would probably be used.

Standard automatic weapon actions generally extract and feed by recoil or gas pressure. It was not deemed desirable to extract the primer while pressure remained in the bore. Also, a primer should not be fed into the firing position until just before firing the cannon. The system also should be activated by the motion of the breech rather than be externally powered.

To meet the above criteria, it appeared that a straight line locking arrangement would be in order to simplify the system and take advantage of the sliding breech motion. The initial design used a basic rotating bolt type lock. However, this was activated by a straight forward and aft motion of the bolt carrier (see Appendix D, Layouts D-1 and D-2). This system had the disadvantage of length, complexity, and probably the necessity to round the forward end of the primer to permit smooth feeding into the chamber.

The concept chosen (see Appendix C, Drawing C-2) is a simple mechanism activated by a cam surface on the breech, which provides a straight rearward pull on the bolt lever. As the bolt carrier moves to the rear, the firing pin is retracted, releasing the locking lugs. The bolt comes forward, driven by a cam on the breech, as the breech closes. This feeds a primer from the basic 10-round clip or optional 30- to 60-round drum (see Appendix D, Layout D-3). The primer is positively guided into the chamber and is engaged by the extractor. When fired, the bolt is driven forward by a spring developing approximately 60 pounds force. The firing pin reacts directly on the locking lugs. The forward motion of the firing pin moves the locking lugs out into position and fires the primer. If the bolt is not locked or the lugs are unable to lock for any reason, the primer cannot be fired. During the breech opening, the bolt carrier moves to the rear with the firing pin. The locking lugs are released and the spent primer extracted and ejected. The firing pin locks in the rear,

or cocked, position. The entire assembly is mounted in an interrupted thread housing which allows easy, fast removal from the weapon for service or replacement. The primer may be fired using the integral solenoid or manually with a lanyard. (See Appendix C, Drawing C-2 and Appendix D, Layout D-3.)

All components of the primer feeder are rugged, minimum tolerance parts for reliability and ease of maintenance.

2.2.4 Controls

The autoloader controls (see Figure 2-5) present a go/no-go display for the gunner and/or assistant gunner. Standard solid state circuitry techniques will be used (see Figure 2-6). The controls are capable of fully automated operation when interfaced with a fire control computer system or they can operate in an autonomous fashion. Three modes of operation are available. The manual mode requires the gunner to address each function and activate it as well. The auto mode requires the gunner to select rate, projectile, charge, and number of rounds only. All other functions except the fire initiate are done automatically. In the computer mode, the system is interfaced with the fire control system and reacts to the fire order sent from the FDC. The gunner commands by exception and can hold or abort the mission from his controls. In the use of the computer controlled system, the FDC could have a similar control panel or a reduced function panel to monitor the system functions and maintain ultimate control of each weapon system. Should a problem occur during operation of the system, it will show a red light and stop at that function until corrective action is taken. Each operation is fail safe and must be in a "go" mode for the system to advance to the next position. During the automatic operation, the go/no-go signal will appear as the function is performed. The fire order

(Not to Scale)

FUNCTION							Fail	Ready													
MODE	MANUAL	AUTO		COMPUTER			RED	GREEN													
RATE	SINGLE	SUSTAINED		BURST			R	G													
NUMBER ROUNDS	1	2	3	4	5	6	R	G													
PROJECTILE	HE	WP	A	I	F	NUMBER REMAINING	R	G													
CHARGE	3	5	7	8	9		R	G													
FEEDER	ACTIVATE	HOLD		SELECTED PROJECTILE AND CHARGE IN POSITION			R	G													
LOADER	ACTIVATE	HOLD		PROJECTILE AND CHARGE READY TO RAM			R	G													
RAMMER	ACTIVATE	HOLD		PROJECTILE AND CHARGE RAMMED-NO FALLBACK			R	G													
BREECH	CLOSE	OPEN		HOLD			R	G													
PRIMER	CLOSE	OPEN		HOLD			R	G													
RECOIL SYSTEM - FAIL SIGNAL REQUIRES CHECK OF RECOIL PANEL							R	G													
COORDINATES	FIRE ORDER						R	G													
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FIRE	GUN POSITION AZIMUTH						R	G													
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FIRE	HOLD																				
150°																					
	ABORT																				

Figure 2-5. Autoloader Panel

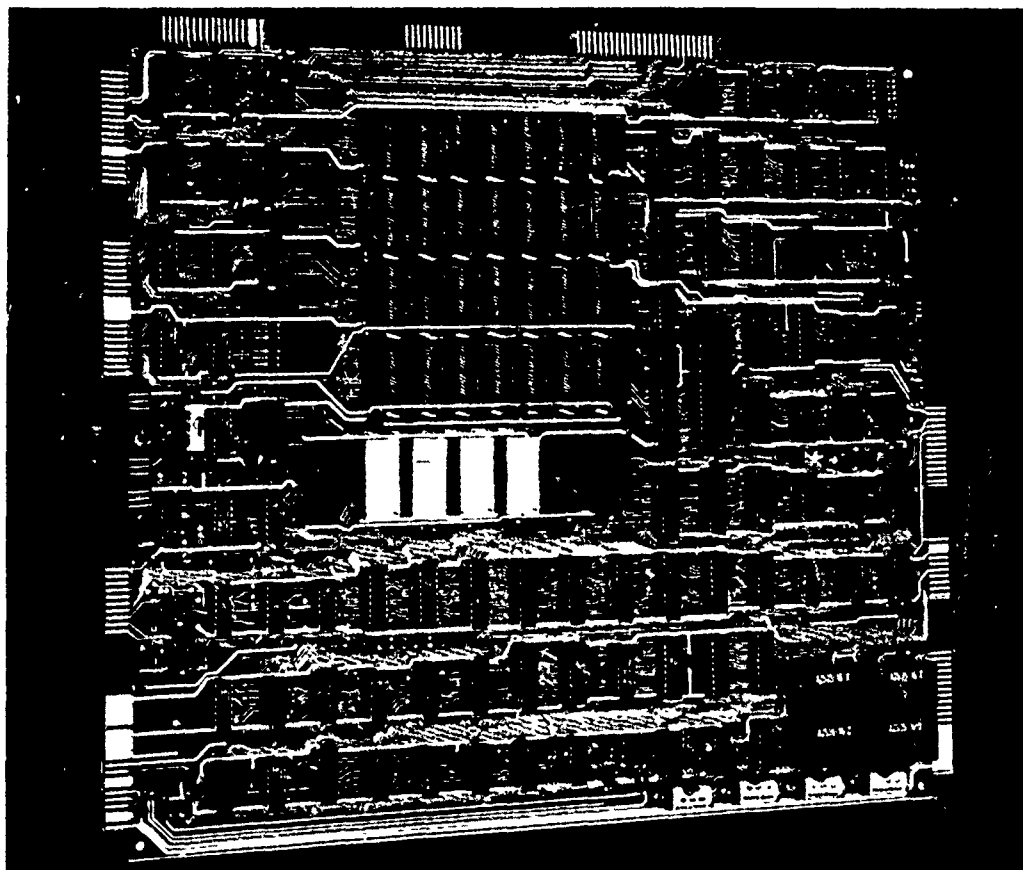


Figure 2-6. Standard Type Programmable Controller

coordinates are displayed in the upper panels, and the system will not advance to the fire mode until these numbers are matched either manually or by the gun system. The go/no-go light for the recoil system will indicate if a problem exists, thereby, notifying the gunner to check the recoil system panel for the specific problem. The panel could incorporate fallback indication as well as tube temperature and the amount of time the breech has been closed or the safe time to cook-off. The hold button will simply hold the system ready and require the gunner to lift the cover and press the fire button. The abort switch will open the breech and extract the primer. The ammunition select buttons will also serve as an indicator for the number of each type of projectile and charge remaining in the stowage racks. As the gunner selects the type and number of projectiles and powder, the switches will light up, e.g., if he should press the number 4 to indicate 4 rounds, and then selects HE for the projectile, the number 2 indicator would begin flashing if he only had 2 HE rounds left in the rack. The same would hold true for the charge as well. An added feature would be a digital counter to indicate the number of units remaining for each. To ensure the proper projectile and powder charge will be fed to the weapon, sensors for weight of charge and type of projectile are incorporated on the feed trays. The system will hold and indicate the type of units in the feed trays by flashing the appropriate select buttons.

The use of a simple micro-processor with a memory will allow the unit to be used for other uses such as hatch and spade positions, engine functions, various liquid levels, etc., if desired. If a fire control system is used, the fire control computer could be interfaced with the control sensors, thus, eliminating the need for a separate micro-processor.

2.3 CONCLUSIONS AND GROWTH POTENTIAL

The complete autoloader system offers a growth potential for the 155mm weapon mounted in the M109 vehicle and all other cannons using one- or two-part ammunition desiring the ability to load and fire at all degrees of elevation or while stabilized or moving.

The basic design, as reported here, is the result of a study in a limited time with specific requirements for interface with the existing weapon system. An improved version of the system, as desired for the M109, would include development of some of the following areas.

Additional racks of ammunition can be carried in the area forward of the existing racks using a mechanism to move the ammunition rearward to be accepted by the existing carriage. Either of these systems would provide up to 50 or 60 ready rounds in the existing vehicle. The vehicle would still maintain enough space to carry an adequate supply of Copperhead and special purpose rounds.

A study to determine the feasibility of an underslung carriage to accept the projectiles and powder charges directly from the elevator would provide a simple solution for the addition of another rack of ammunition forward of the existing racks.

The compact nature of the autoloader and storage racks will permit isolation of the gunner and assistant gunner by providing an enclosure that will turn with the turret, with doors into the gun compartment as well as hatches in the turret roof. This arrangement would provide smoke, heat, noise, and CBR protection for both crewmen. The system should be capable of manual operation and maintenance by the gunner and/or assistant gunner, eliminating the need for additional crewmen. Should sustained firing be required, additional crewmen would be assigned. The autoloader system as designed could provide

total autonomous operation of the M109 when interfaced with a fire control system.

The telescoping arms of the autoloader could possibly be replaced by a strong back chain, over which the carriage could travel. This would eliminate some complexity of the system.

A compromise of the turrets ability to traverse through approximately 120 degrees rather than 360 degrees could provide a lower profile with increased stowage of ammunition because the bustle would extend much lower and turn with the turret.

The anticipated 5 seconds between rounds might be further optimized, depending on the weapon configuration and recoil system.

3.0 RECOIL SYSTEM

3.1 RATIONALE

The basic goal of this effort was to design a recoil system that could be retrofitted to the M109 and that would provide a significant improvement in reliability and maintainability over the existing system. This was achieved. The integral buffers and replenishers which eliminate all hydraulic plumbing will provide this significant improvement.

A secondary goal to provide safe operation if one cylinder of either pair of recoil or counterrecoil cylinders becomes inoperative was not achieved. This is due to the following two reasons.

First, to permit a reasonable retrofit, only one counterrecoil cylinder can be utilized. The location where a second cylinder could be installed is occupied by the direct fire telescope. There seems to be no practical way to relocate this telescope in a retrofit program.

Second, the impulse of the rounds expected to be fired from the new gun is so great that it became impractical to consider permitting firing with only one recoil cylinder operating.

Modification of the gun mount structure required to accept the new recoil system consists basically of cutting off and boring out the existing welded-in recoil cylinders. The new gun mount assembly will be very "clean". The buffer cylinder, replenisher cylinder, and all hydraulic tubing and fittings will be gone.

A constant length recoil system was selected because analysis showed that recoil length had little effect on vehicle motion resulting from firing. (See Appendix B3.) Vehicle motion is primarily a function of the magnitude of the impulse of the round fired. It is effected only slightly by

changes in trunnion reaction, unless the recoil length can be made long enough so that static equilibrium is approached. Such a long recoil travel is not practical in the M109.

Although the new gun and ammunition have not been finalized, the following weapon characteristics were furnished by ARRADCOM for design purposes:

Projectile weight.	98 lb
Projectile velocity.	3,250 ft/sec
Propellant weight.	40 lb
Recoiling weight	9,600 lb

3.2 DESIGN (see Appendix B1 and Appendix C, Drawings C-3, Sheets 1 and 2)

3.2.1 Recoil Cylinder

The following is a discussion of the major features of the recoil cylinder.

3.2.1.1 Rod Seal

The rod seal is a unique feature of the recoil cylinder design. The seal is not subjected to the high pressure (over 6,000 psi) during recoil. The high pressure is reduced to almost zero by the labyrinth grooves and is then bled off to the low pressure end of the cylinder. The rod seal will be subjected to only the pressure required to move the oil that leaks through the labyrinth grooves to the front end of the cylinder. During recoil, the front end of the cylinder is actually under a vacuum caused by the displacement of the piston rod.

3.2.1.2 Buffer

The buffer is a 1-3/4 inch diameter, 6-inch-long spear which plugs into a cavity in the piston rod. The spear has three parabolic shaped orifice grooves designed to bring the weapon to a stop with a constant force acting over a 6-inch travel. The buffer is absolutely foolproof since it has no moving parts. Also, during counterrecoil, oil is forced from the front end of the cylinder to the rear end, transferring the vacuum from front to rear, thereby, assuring the buffer cavity is full of oil.

3.2.1.3 Orifice Sleeve

For a fixed-length recoil system, the orifice sleeve has a number of advantages over a control rod. The piston, piston rod, and buffer are much simpler. In addition, the sleeve provides a means of piping the bleed oil from the labyrinth seal in the rod gland to the other end of the cylinder without external plumbing.

3.2.1.4 Replenisher

Both recoil cylinders are equipped with an integral replensihier. The replenishers have a nitrogen spring and are at sufficient capacity to accommodate all operational temperatures and provide a maintenance-free life of over 10,000 rounds.

Electronic sensors are installed in the replenisher to indicate the status of the oil volume, which is displayed as estimated rounds before maintenance on a display panel.

3.2.2 Counterrecoil Cylinder

The existing M109 counterrecoil cylinder is completely self-contained,

and this basic design has been retained. However, two major modifications have been made to the cylinder:

- The cylinder has been shortened by approximately 16 inches to take advantage of the fixed 21-inch recoil travel.
- Electronic sensors have been included to provide a readout on a display panel of the conditions of the rod seal and piston seal.

3.2.3 Controls



The Recoil System digital readout panel (see Figure 3-1) can be mounted either at the gunner's station or at both the gunner's and assistant gunner's station. Each unit indicates the condition of the recoil system by a percentage readout. If any function reaches its lower limit, an X will begin flashing in the numeral 1 position of that indicator. Simultaneously, the fire inhibit circuit will be activated, interrupting the trigger switch. Therefore, to fire the weapon, the gunner will be required to engage the override switch on the panel, thus, acknowledging a lower limit condition within the system. The orange displays can be quickly and easily read in direct sunlight and at a distance of 20 feet. This will allow the displays to be monitored while recharging.

3.3 CONCLUSIONS AND GROWTH POTENTIAL

The recoil system discussed here is a significant improvement for the M109A2 because it is a simple design that will provide improved performance, it is designed for RAM-D improvement, and it enhances performance of the autoloader. As explained earlier, the recommended approach has traded-off the redundancy possible with a four-cylinder system in order to make

**RECOIL SYSTEM DIGITAL READOUT PANEL
(SCALE: FULL SIZE)**

BECKMAN SP330 DISPLAYS

RECOIL FUNCTIONS	LOWER RECOIL	UPPER RECOIL	COUNTER RECOIL	COUNTER RECOIL FUNCTIONS
REPLENISHER PRESSURE 25 PSI	PRESSURE % 100 LOW LIMIT 10	PRESSURE % 100 LOW LIMIT 10	PRESSURE % 100 LOW LIMIT 1500	CYLINDER PRESSURE 2000 PSI
REPLENISHER OIL VOLUME	PISTON OIL % 100 LOW LIMIT 5	PISTON OIL % 100 LOW LIMIT 5	PISTON OIL % 100 LOW LIMIT 5	PISTON OIL VOLUME
FIRE INHIBIT SWITCH WITH COMBAT OVERRIDE	FIRING SWITCH OVERRIDE  INHIBIT	TEST  ALL UNITS 100%	ROD OIL % 100 LOW LIMIT 5	ROD OIL VOLUME

TEST SWITCH
MOMENTARY ON, ALL
INDICATORS READ 100%

Figure 3-1. Recoil System Digital Readout Panel (Scale: Full Size)

possible a low-cost adaptation to the M109A2.

Future design of an all-new gun and turret combination can use paired recoil and counterrecoil cylinders. The potential safety and life advantages of such a system would then have to be traded-off against the extra weight and cost.

4.0 RAM-D

4.1 AUTOLOADER R&M MODEL

This report presents a preliminary assessment of reliability and maintainability characteristics of an automatic feed, load, and recoil system concept for a 155mm self-propelled howitzer. The data presented are considered to represent typical values for generic components in a severe environment. While the assessed values for individual components likely possess a large degree of inherent error, both high and low, some errors should cancel, and the result indicates a ballpark figure for the total assembly. In this regard, the analysis should not be considered as conclusive, but rather as a point of departure to stimulate thinking, arouse concerns, and guide follow-on efforts.

The three values presented on the block diagrams are defined below:

- λ_F = Failures/ 10^6 rounds

Where failure results in complete loss of autoloader capability.

No allowance is made for the capability to revert to manual loading.

- λ_m = Maintenance actions/ 10^6 rounds

Where maintenance action is considered as any repair necessary to retain full capability. This includes replacement of failed components, adjustments, and preventive repair (tighten bolts, replace seats, etc.)

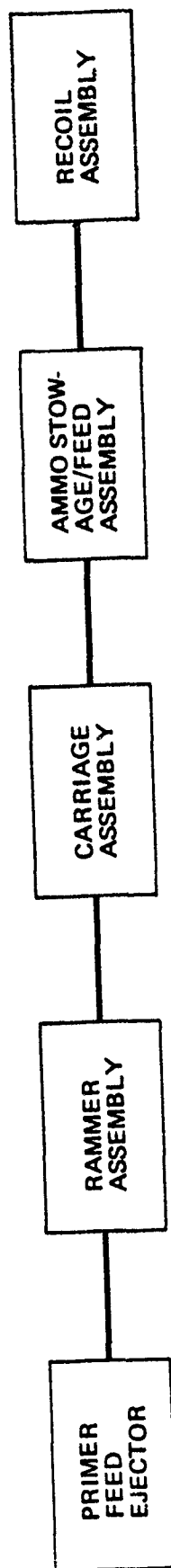
- μ_r = Average corrective maintenance time in hours for maintenance actions. This includes only action immediate to the autoloader/recoil assembly and assumes repair parts are readily available.

$$\text{MRBF} = \text{Mean Rounds Between Failure} = \frac{1}{\lambda_F}$$

$$\text{MRBMA} = \text{Mean Rounds Between Maintenance Action} = \frac{1}{\lambda_m}$$

Any discrepancy in terminology between this portion of the report and other sections, the other sections will prevail.

The R&M Model and related data are depicted in Figures 4-1 through 4-5. Figure 4-1 shows the top diagram with its major subassemblies while Figures 4-2 through 4-5 show the details of the major assemblies



$$\lambda F = 10.00$$

$$\lambda M = 150.00$$

$$\lambda F = 22.25$$

$$\lambda M = 83.40$$

$$\lambda F = 9.75$$

$$\lambda M = 43.40$$

$$\lambda F = 34.00$$

$$\lambda M = 128.90$$

$$\lambda F = 6.60$$

$$\lambda M = 31.90$$

$$MRBF = \frac{1}{\Sigma \lambda F} = \frac{1}{82.6 \times 10^{-6}} = 12,107 \text{ ROUNDS}$$

$$MRBMA = \frac{1}{\Sigma \lambda M} = \frac{1}{437.6 \times 10^{-6}} = 2,285 \text{ ROUNDS}$$

Figure 4-1. Autoloader/Recoil System
R&M Model Block Diagram

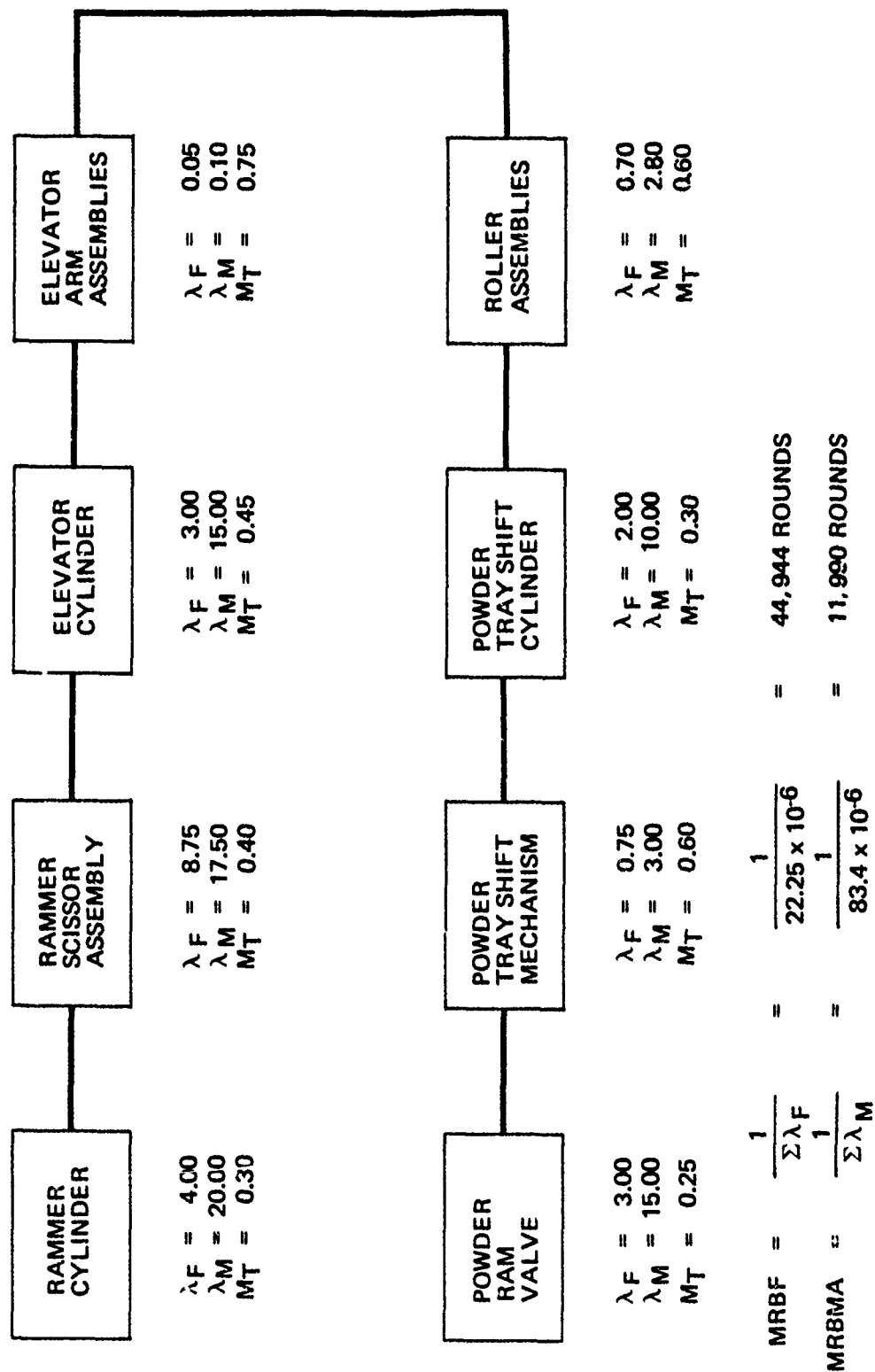


Figure 4-2. Rammer Assembly
R&M Model Block Diagram

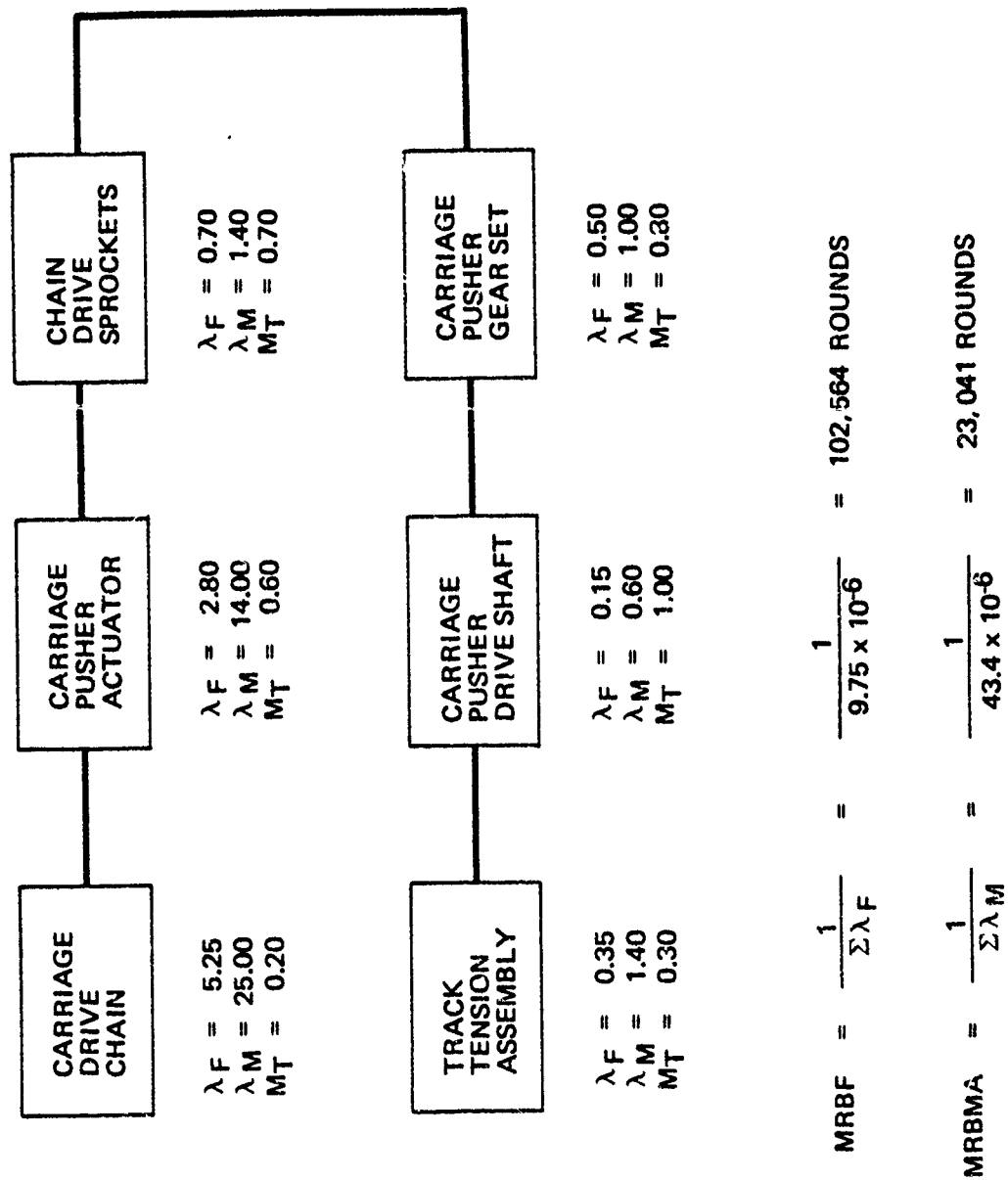
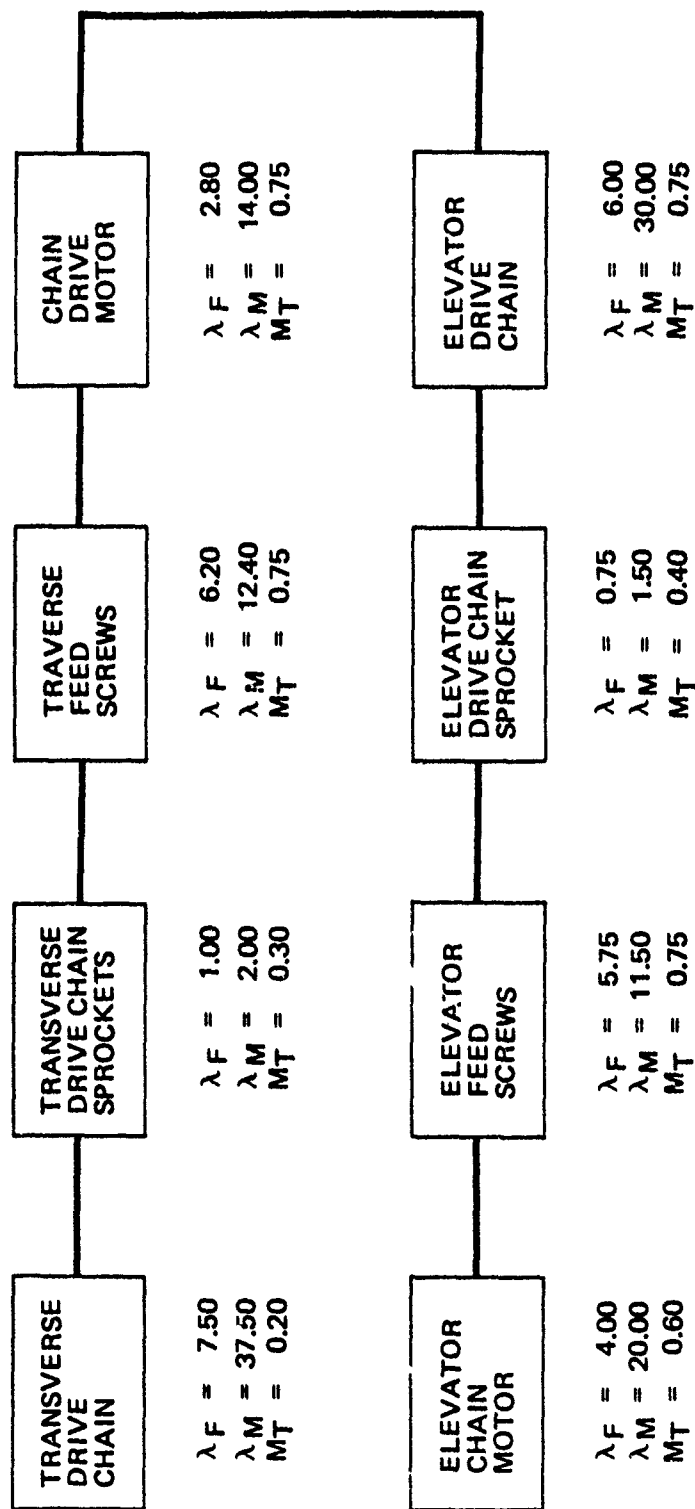


Figure 4-3. Carriage Assembly
R&M Model Block Diagram



$$MRBF = \frac{1}{\sum \lambda F} = \frac{1}{34 \times 10^{-6}} = 29,412 \text{ ROUNDS}$$

$$MRBMA = \frac{1}{\sum \lambda M} = \frac{1}{128.9 \times 10^{-6}} = 7,758 \text{ ROUNDS}$$

Figure 4-4. Ammo Stowage/Feed Assembly R&M Model Block Diagram

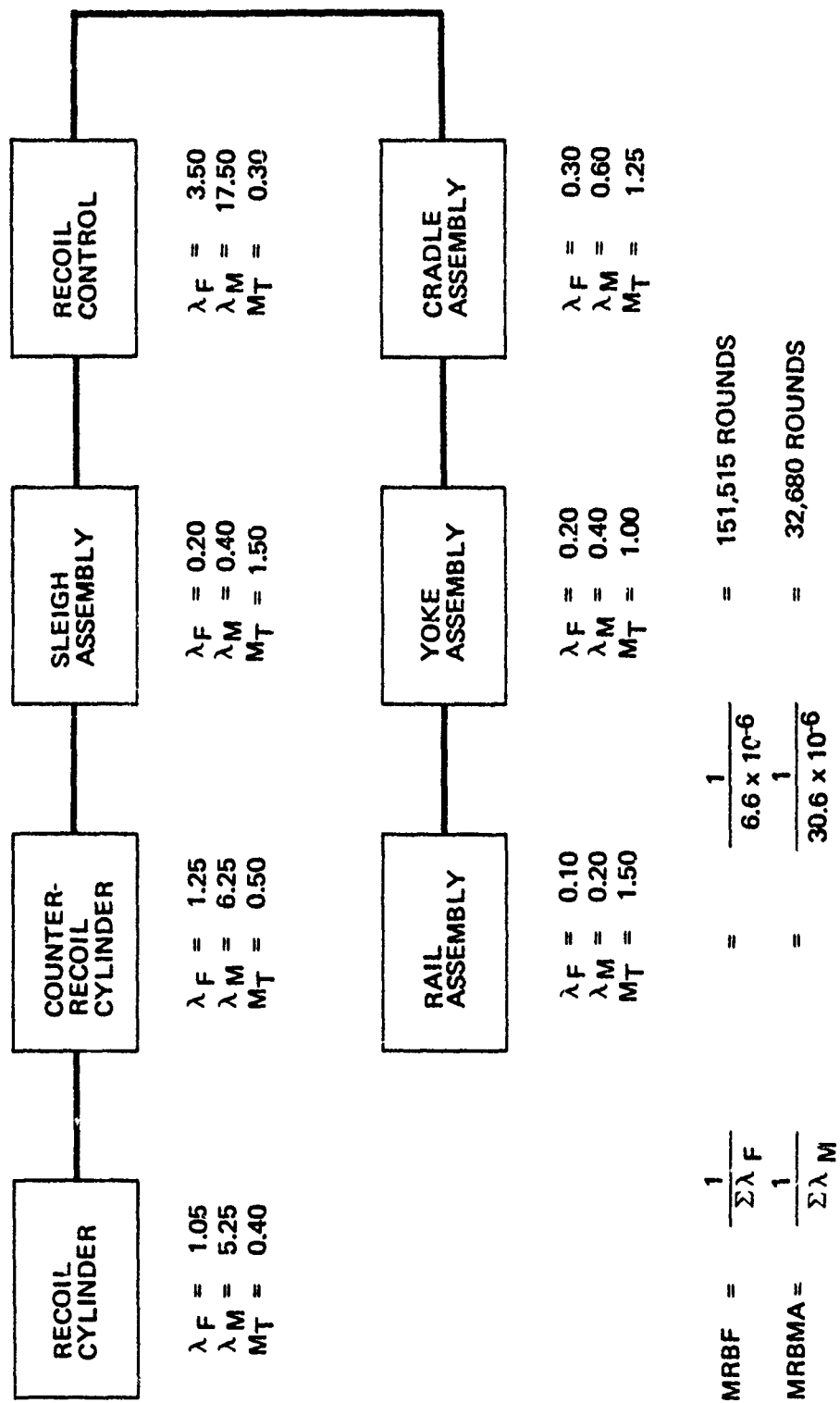


Figure 4-5. Recoil Assembly
R&M Model Block Diagram

APPENDIX A
Autoloader Calculations

PACIFIC CAR AND FOUNDRY COMPANY
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NAME

K. B. E. E.

DATE

2-6-80

REFERENCE

A-1

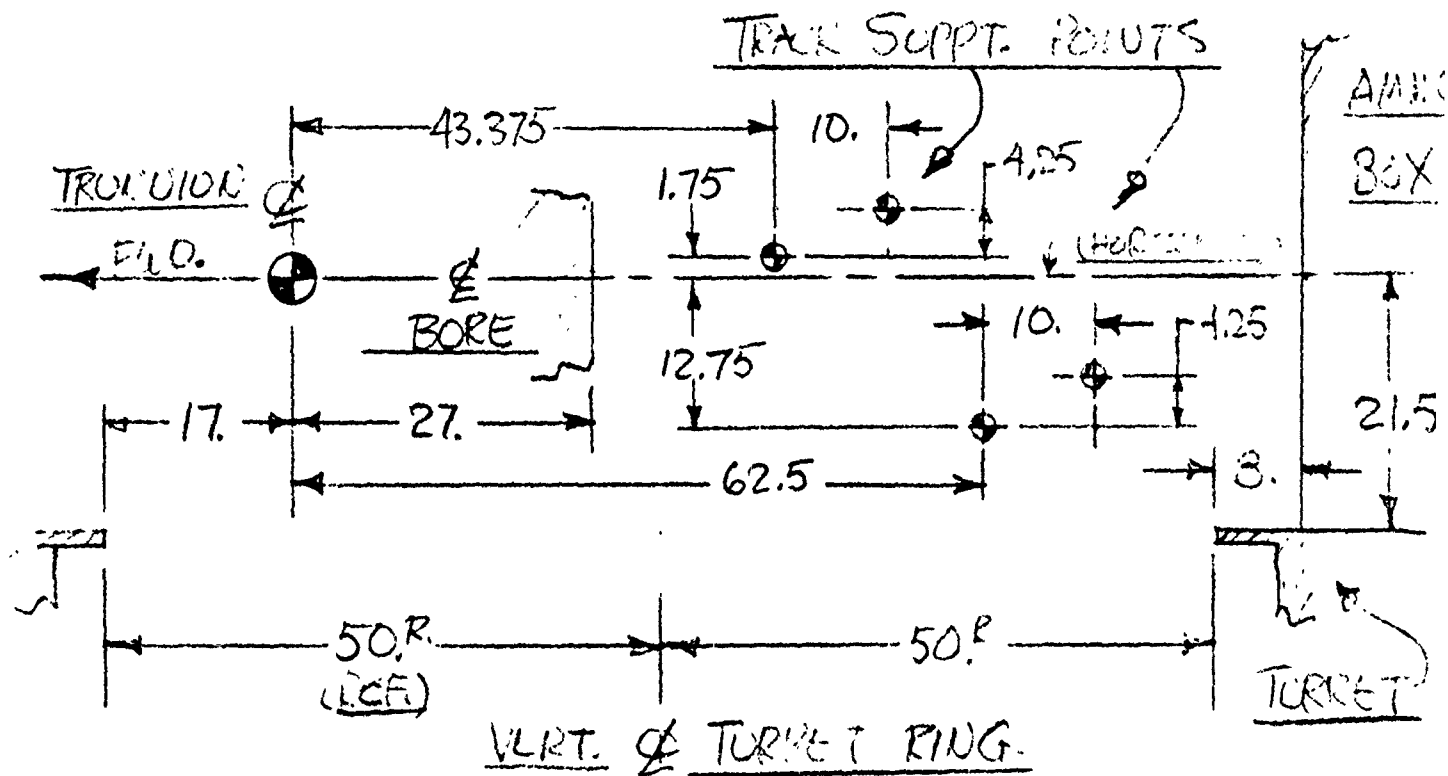
PAGE

1

OF

104

DIMENSIONS



NOTES: THE FWD. TRACK SUPPORTS ARE FIXED TO THE GUN SUPPORT & MOVE WITH THE GUN. THE AFT TRACK SUPPORTS ARE FIXED TO THE AMMO BOX.

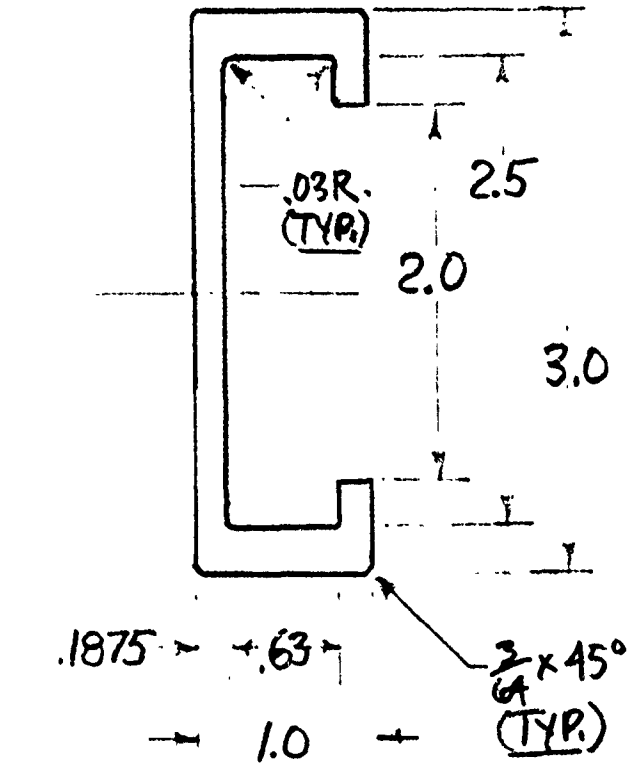
THE RECOIL DISTANCE IS 20 IN.

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME K. BOVEE
DATE 10-29-79

REFERENCE A-L TRACK
PAGE 2 OF

TRACKS



CROSS SECTION (TYP.)

STEEL, STRUCTURAL GRADE

SMOOTH FINISH - NITRIDE

EIGHT (8) PIECES REQD.

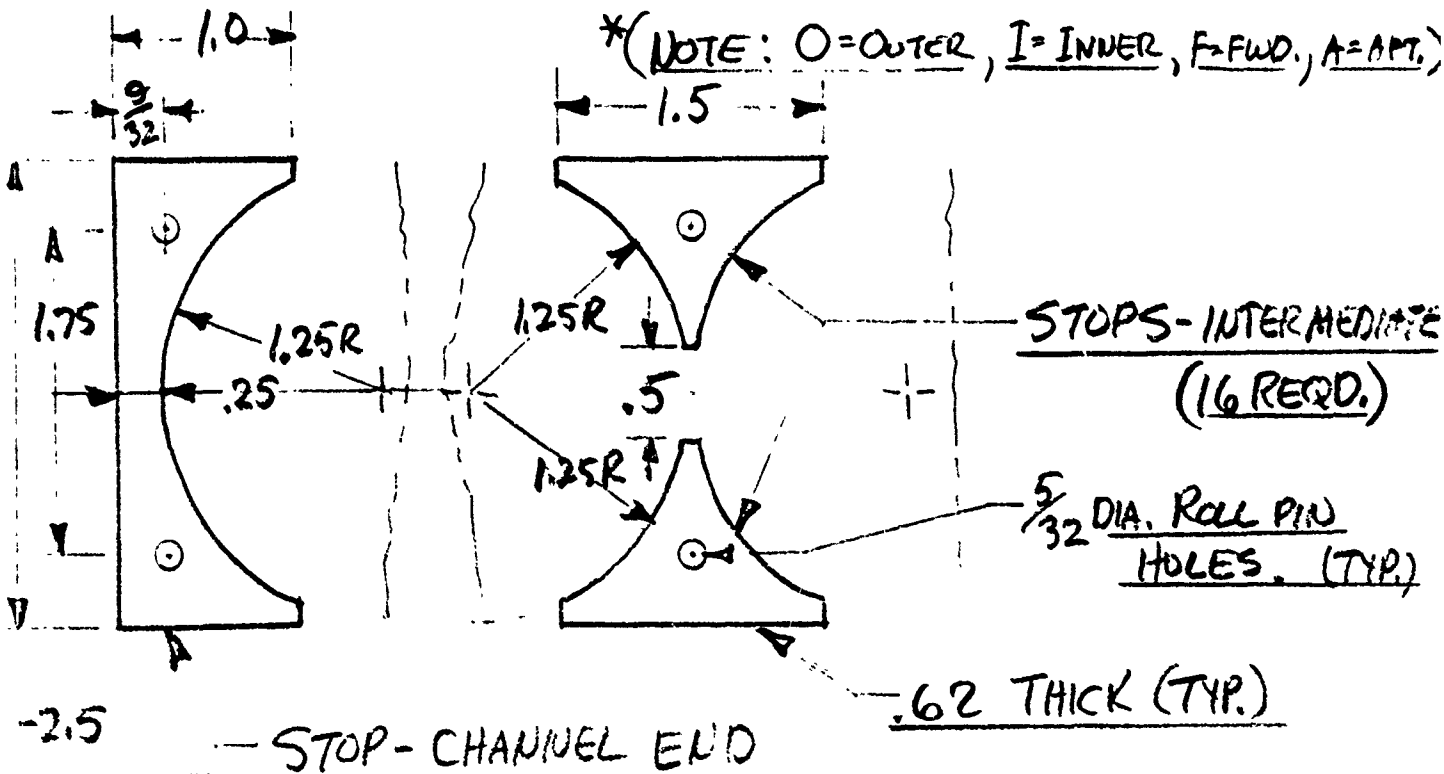
(2) PCS. 36.375" LONG (O*)

(2) PCS. 33.875" LONG (O*)

(2) PCS. 36.25" LONG (I-A)

(2) PCS. 44.125" LONG (I-F)

*(NOTE: O=OUTER, I=INNER, F=FWD., A=APT.)



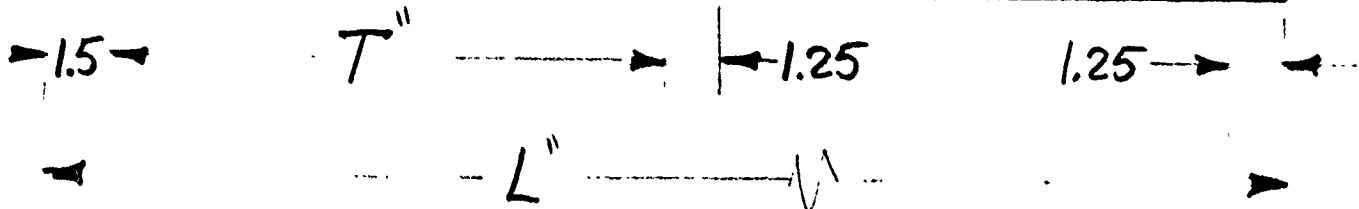
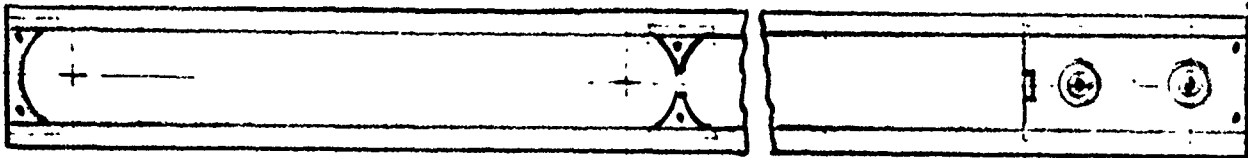
PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME K. B. ...
DATE 10-29-79

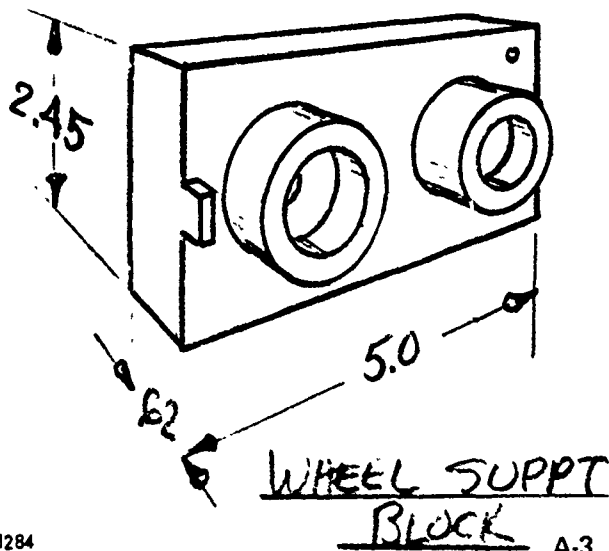
REFERENCE A-L TRACK
PAGE 3 OF

TRACKS

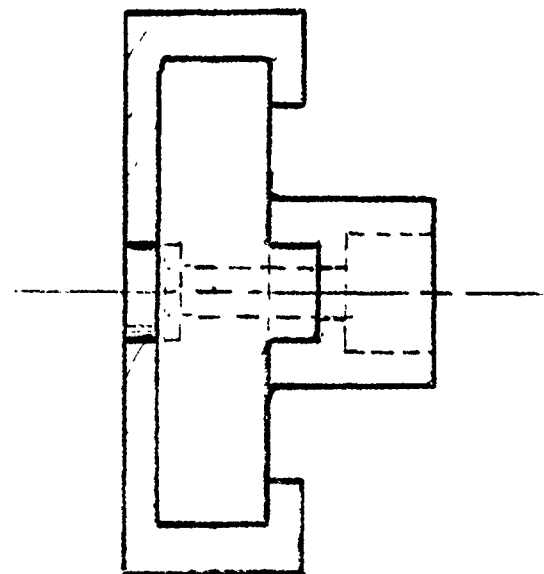
→ 5. →



	<u>T.</u>	<u>L.</u>
1) <u>OUTER FWD. TRACK ~</u>	13 in.	33.875 in
2) <u>OUTER AFT. TRACK ~</u>	15.5	36.375
3) <u>INNER FWD. TRACK ~</u>	24.375	44.125
4) <u>INNER AFT. TRACK ~</u>	16.5	36.25
(T = TRAVEL ; L = LENGTH)		



WHEEL SUPPT
BLOCK



PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

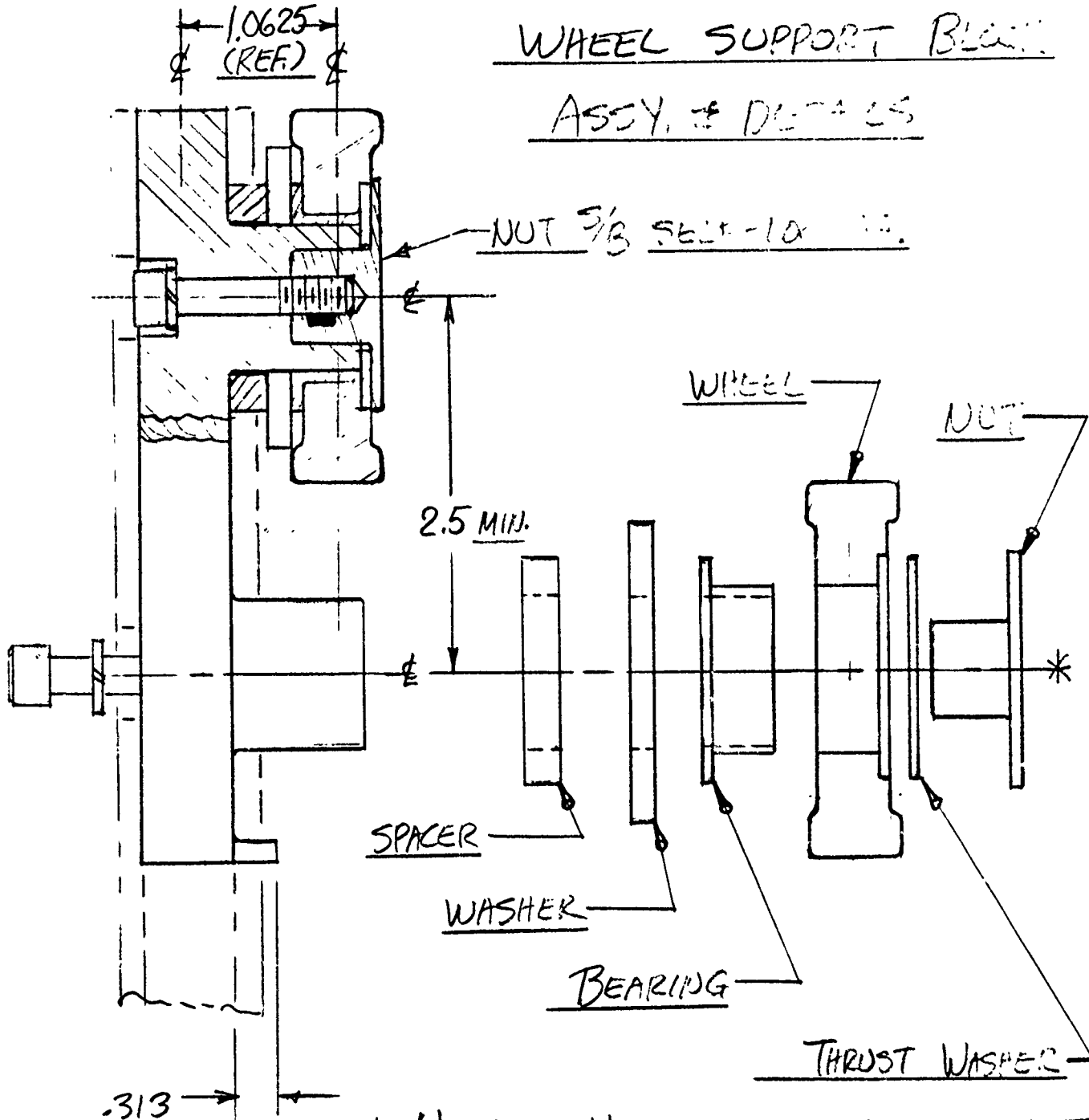
NAME K. BOULE
DATE 10-29-79

REFERENCE A-L TRACK
PAGE 4 OF

TRACK'S

WHEEL SUPPORT BLOCK

ASSY. & DETAILS



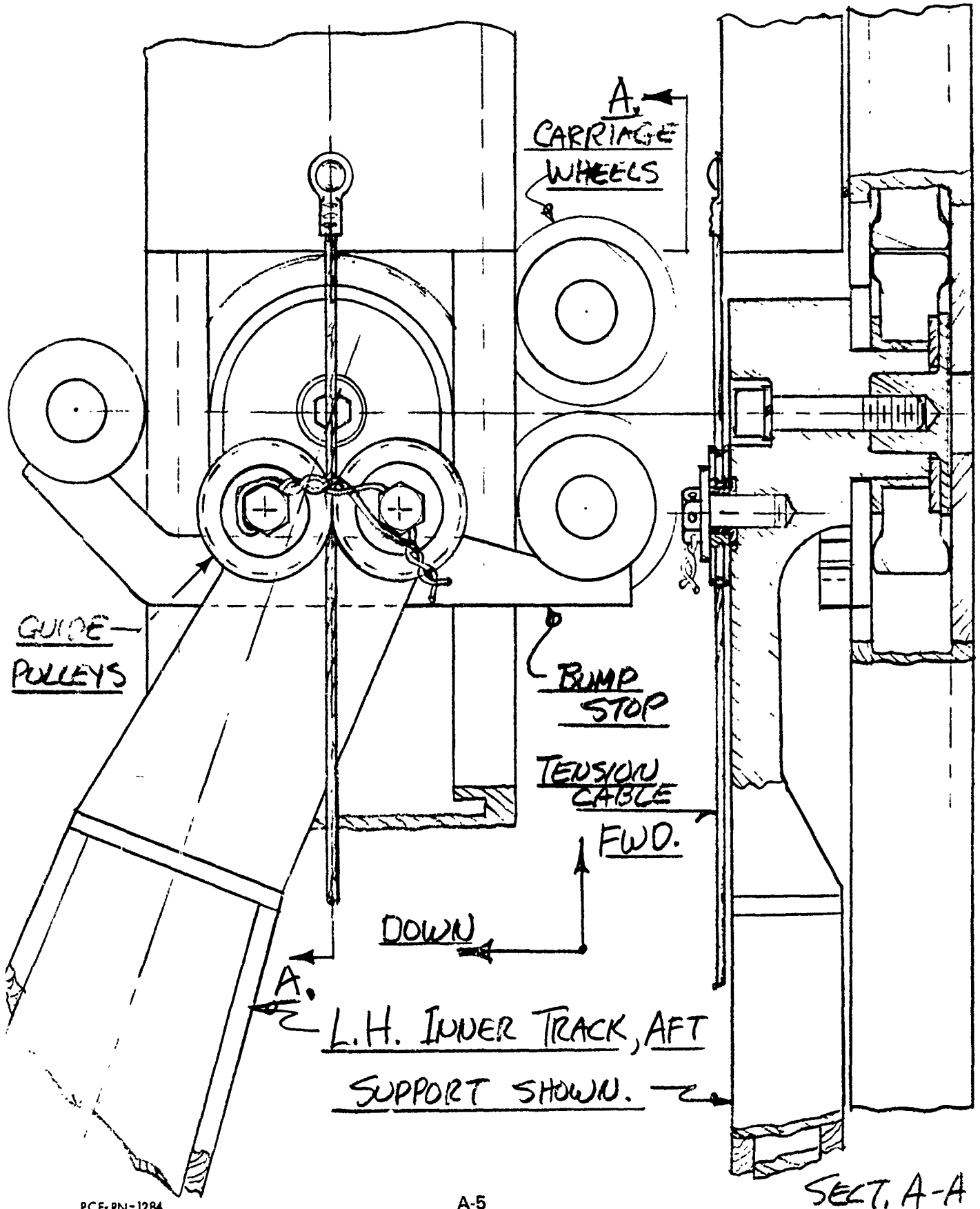
* NOTE: NUT SHOULD HAVE A SLOT
OR EQUIV. IN END SO IT CAN BE

INSERTED FROM END THROUGH HOLE IN TRACK

PACIFIC CAR AND FOUNDRY COMPANY
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NAME H. B. Jones
DATE 10-30-79

REFERENCE A-L TRACK
PAGE 5 OF



PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME

K Bouze

REFERENCE

A-L TRACK

DATE

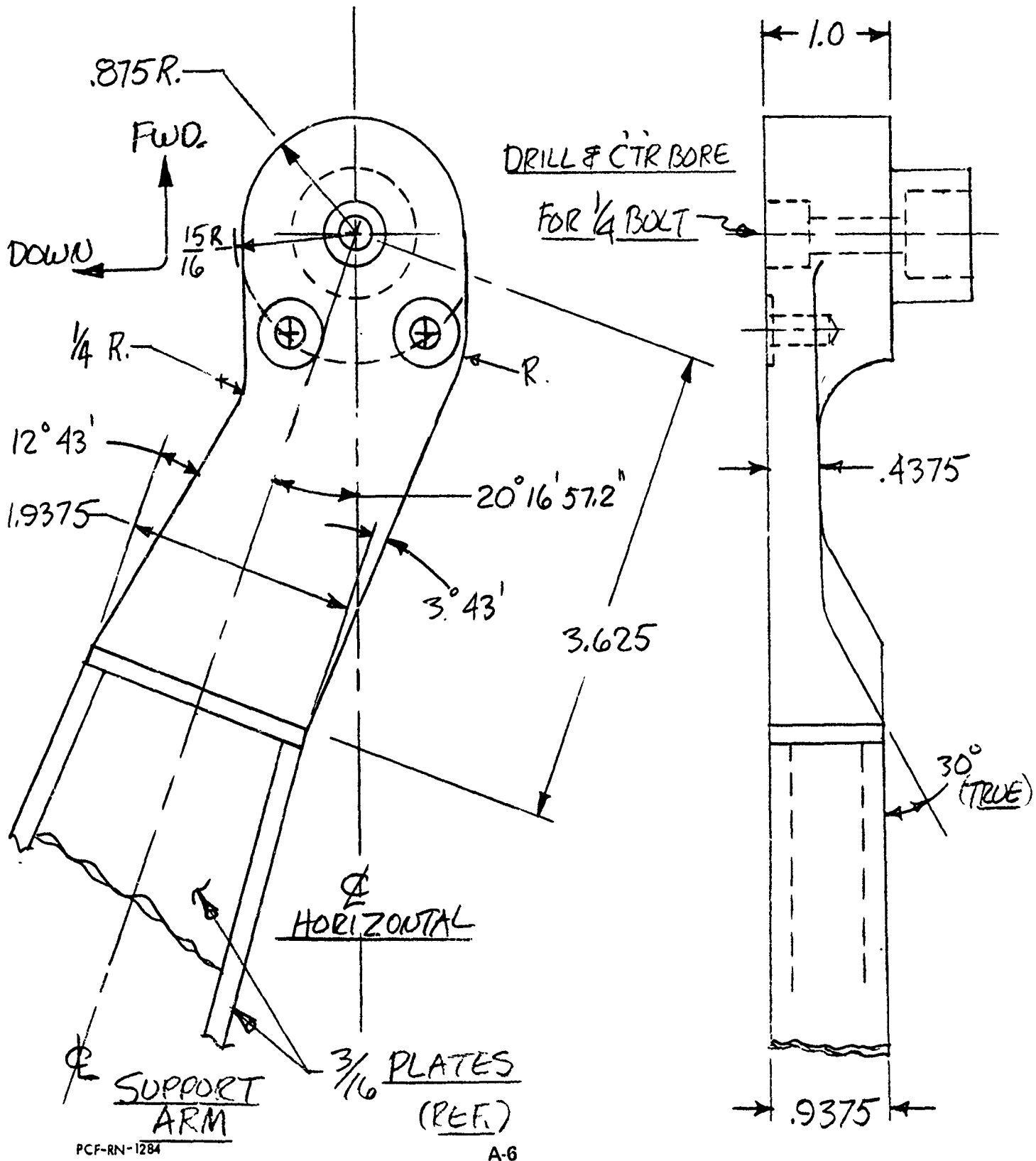
11-1-79

PAGE

6

OF

L.H. INNER TRACK, AFT SUPPORT

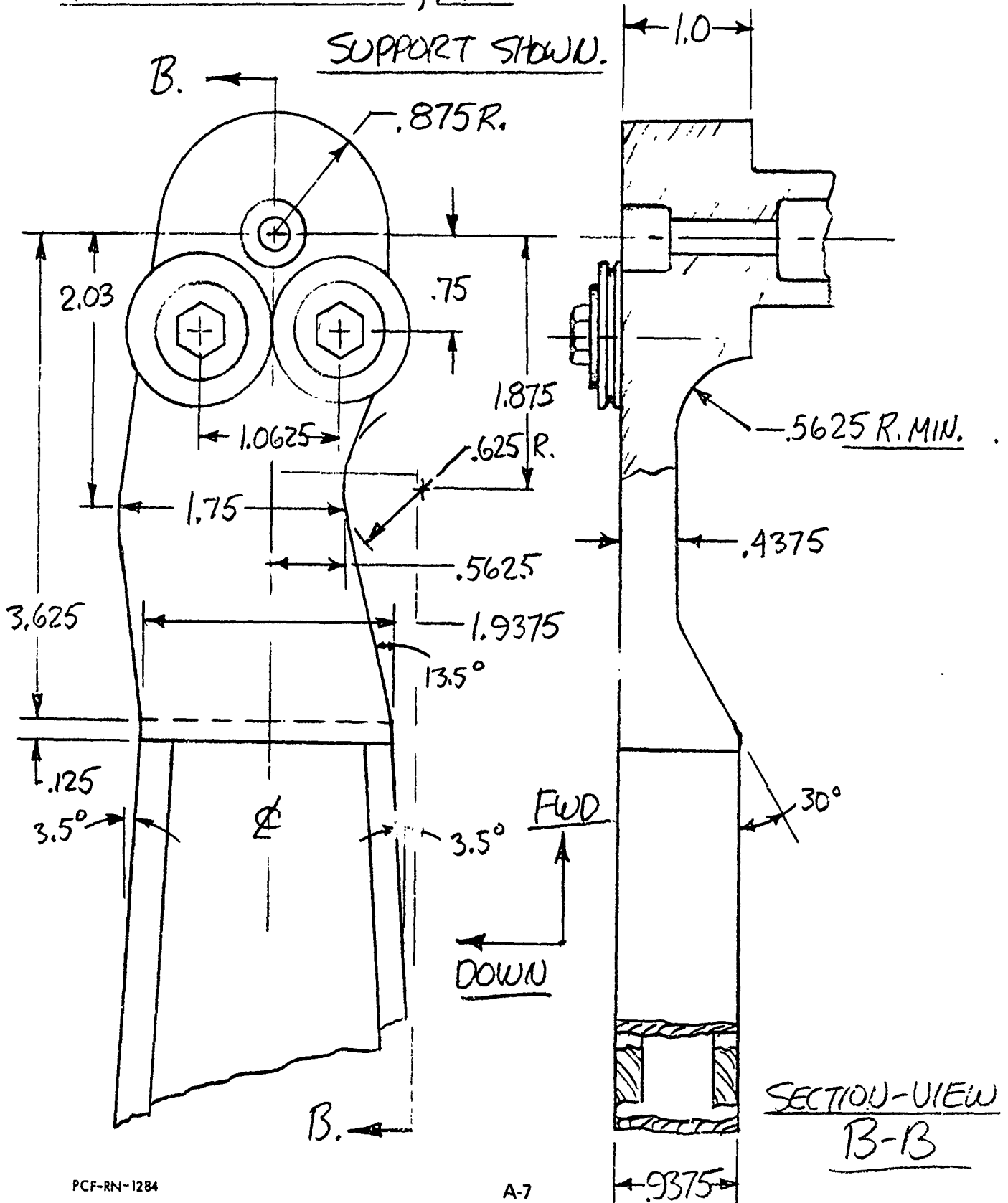


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DATE 10-31-79

REFERENCE A-L TRACK
PAGE 7 OF

R.H. OUTER TRACK, AFT



PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME

K BOVEE

DATE

11-1-79

REFERENCE

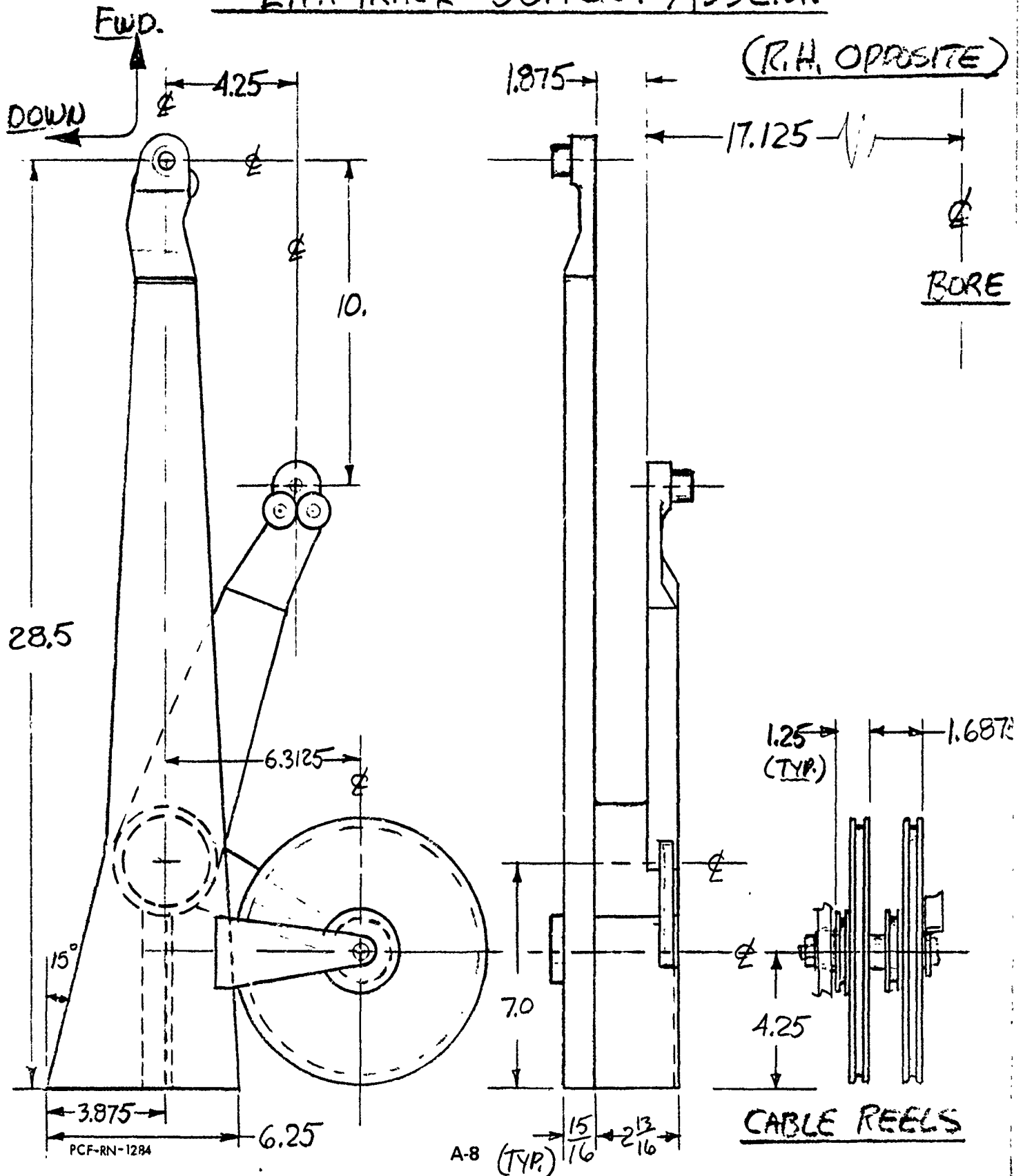
A-L TRACK

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OF

L.H. TRACK SUPPORT ASSEM.

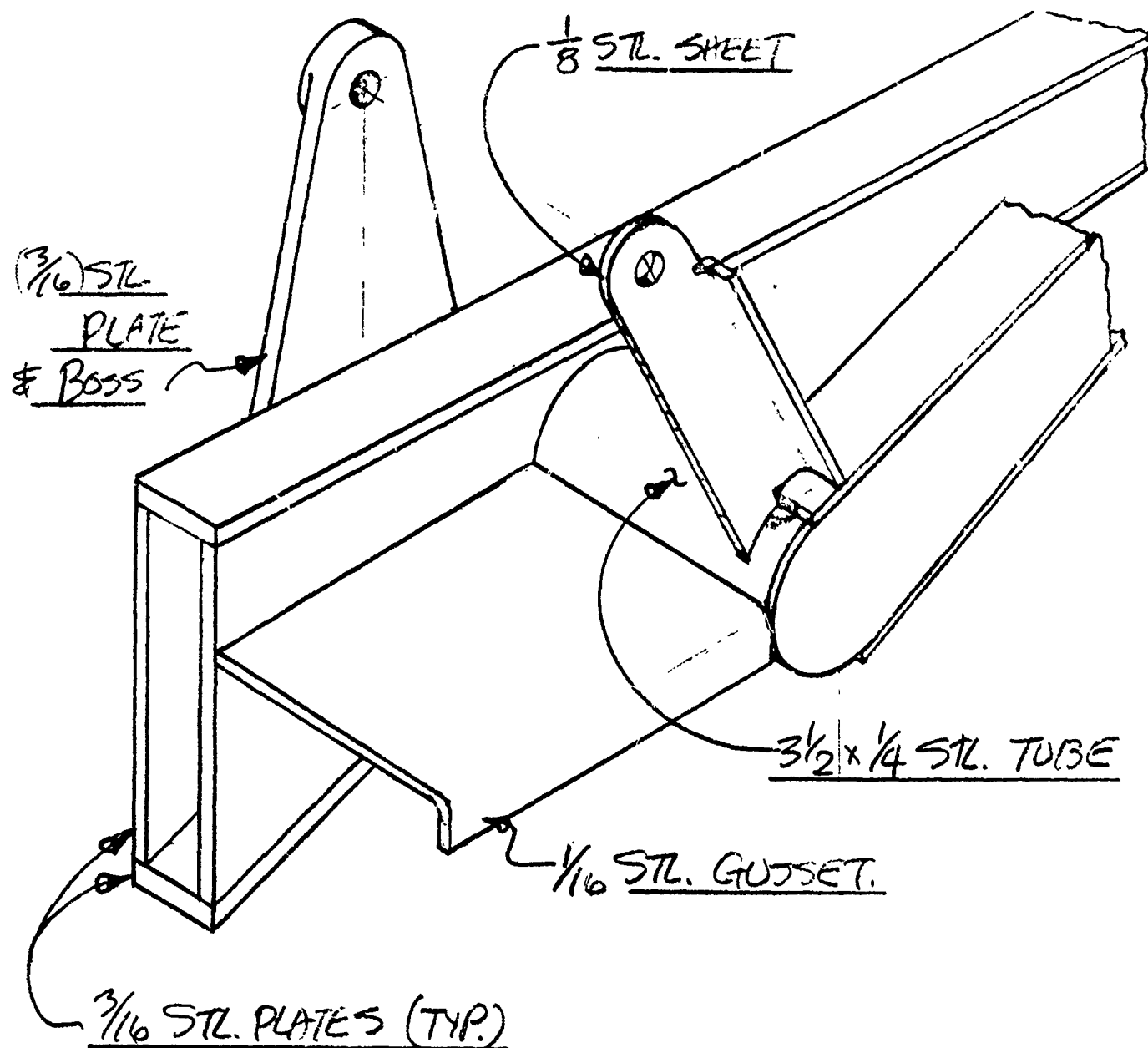


PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME KBOVEZ
DATE 11-2-79

REFERENCE A-L Track
PAGE 9 OF

L.H. AFT TRACK SUPPORT
(ISOMETRIC)



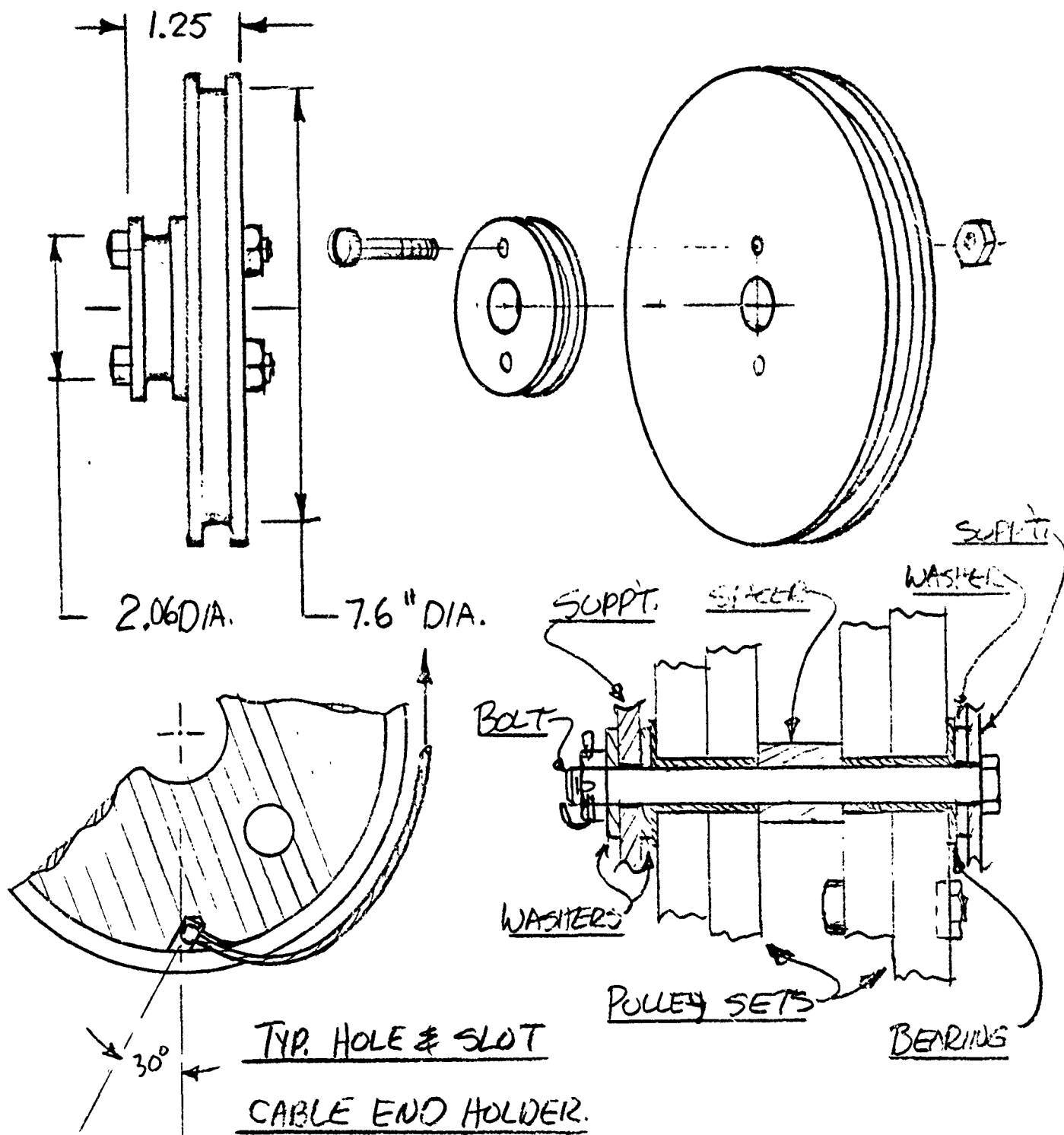
WELDMENT.

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME K. Boone
DATE 11-2-79

REFERENCE A-L. TRACK
PAGE 10 OF

CABLE DRUMS.



POSITION WITH SMALL OVERLAP IN WOUND CONDITION.

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME

K. Boule

REFERENCE

A-L TRACK

DATE

11-5-79

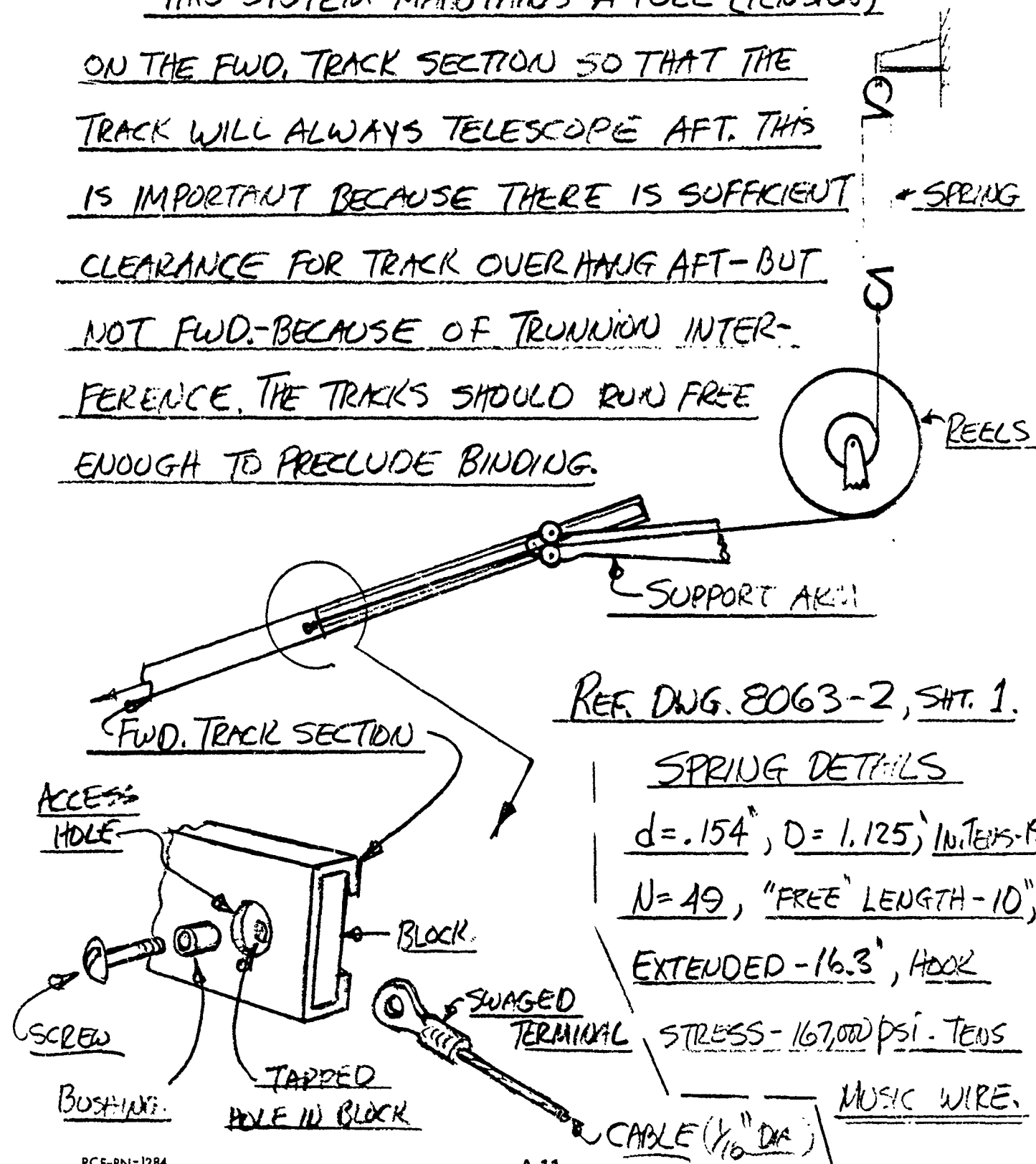
PAGE

11

OF

CABLES & SPRINGS.

THIS SYSTEM MAINTAINS A PULL (TENSION)
ON THE FWD. TRACK SECTION SO THAT THE
TRACK WILL ALWAYS TELESCOPE AFT. THIS
IS IMPORTANT BECAUSE THERE IS SUFFICIENT
CLEARANCE FOR TRACK OVERHANG AFT-BUT
NOT FWD.-BECAUSE OF TRUNNION INTER-
FERENCE. THE TRACKS SHOULD RUN FREE
ENOUGH TO PRECLUDE BINDING.

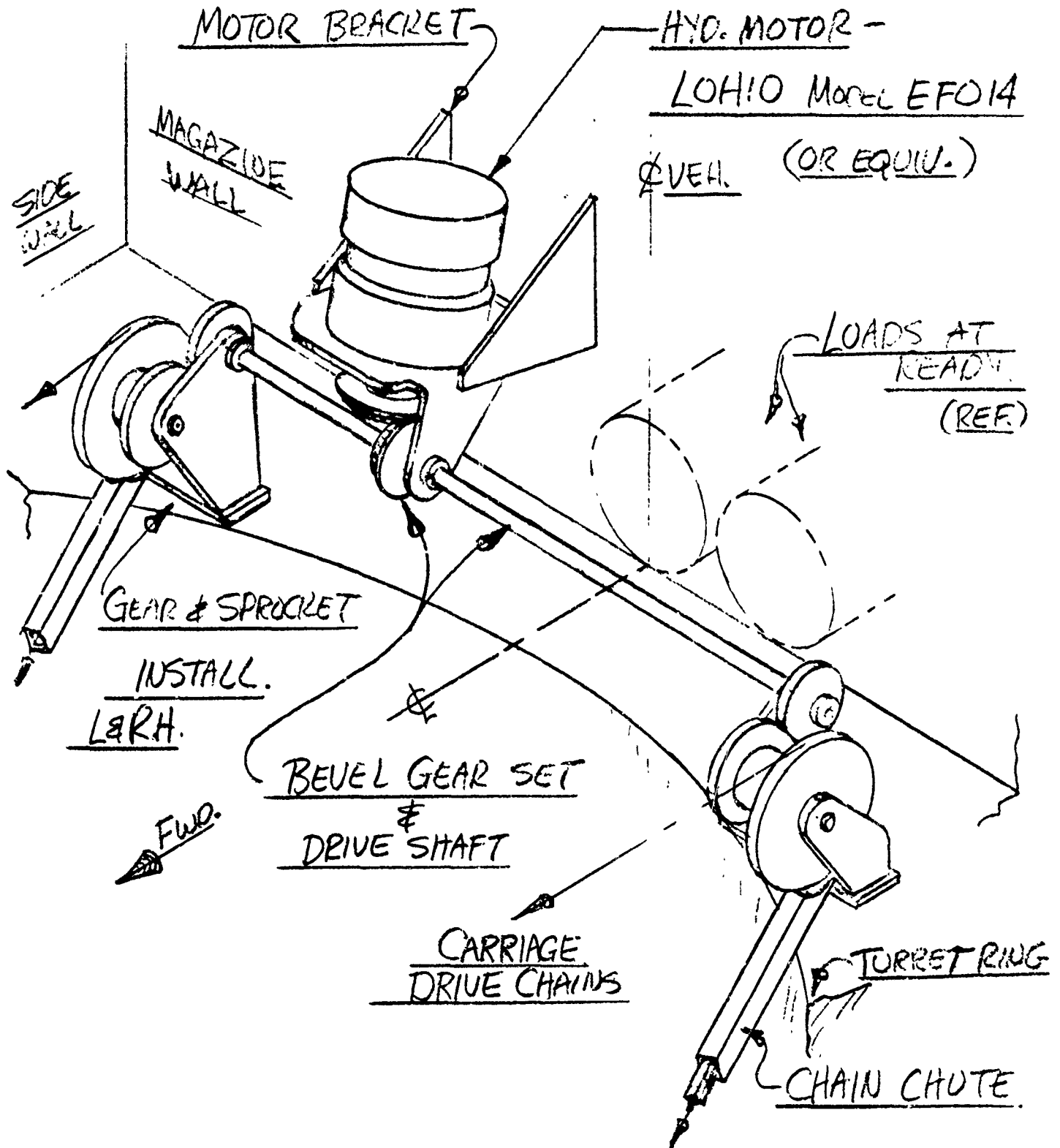


PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME K BOVEE
DATE 11-5-79

REFERENCE A-L
PAGE 12 OF

CARRIAGE PUSHER



PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME

K. BOWE

REFERENCE

A-L

DATE

11-6-79

PAGE

13

OF

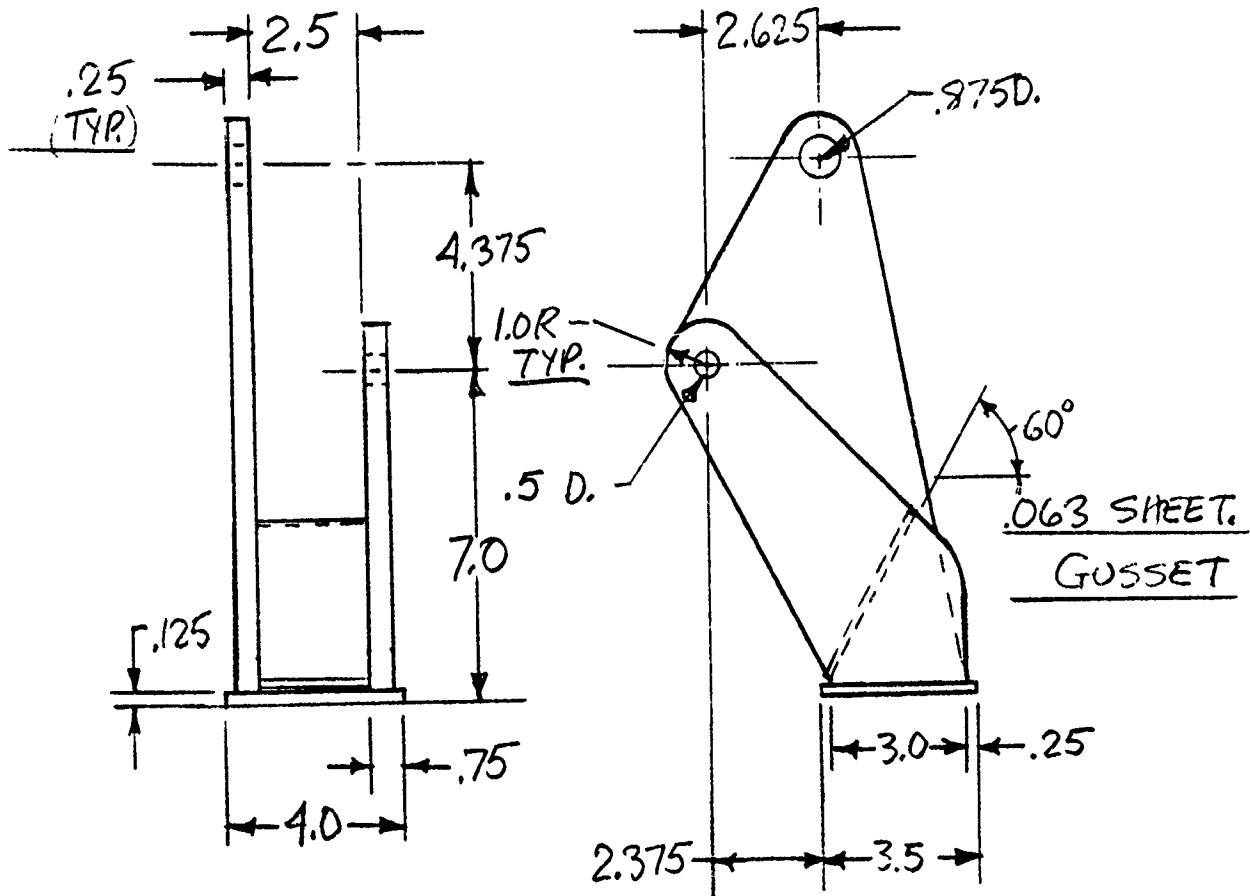
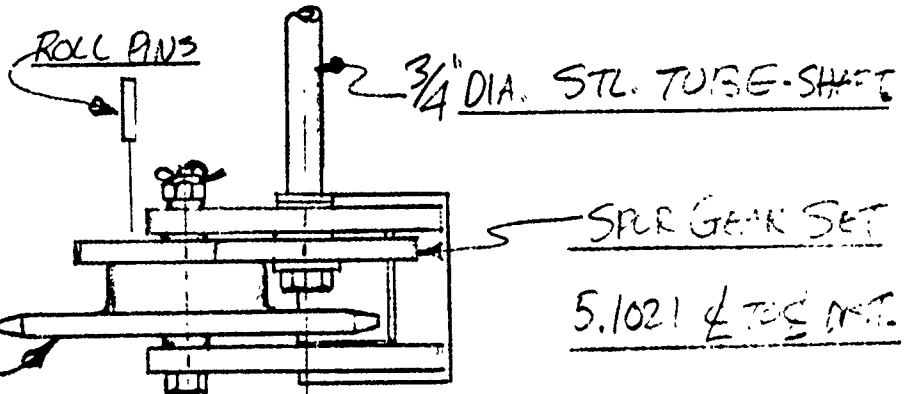
CARRIAGE PUSHER

SPROCKET:

LINE BELT-TYPE B,

34 TEETH, P.D. 8.163

O.D. 8.543, THICK. = .459"



BRACKET ASSY.-WELDMENT

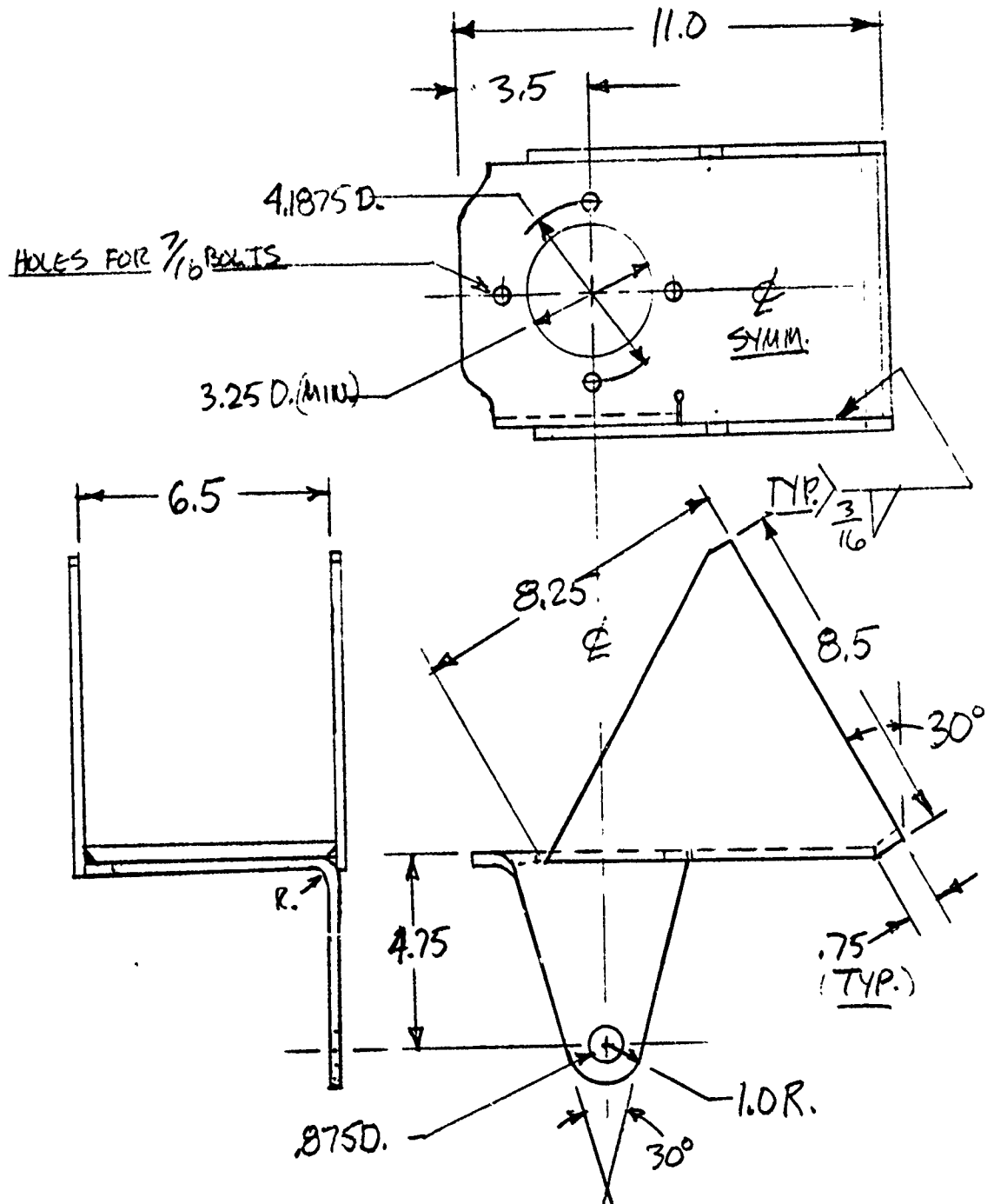
L & R.H. - L.H. SHOWN.

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NAME K. Bovee
DATE 11-6-79

REFERENCE A-L
PAGE 14 OF

MOTOR MOUNT



BKRT. WELDMENT. WELD TO MAGAZINE WALL.

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME

KBouee

REFERENCE

A-L

DATE

1-23-80

PAGE

15

OF

CARRIAGE PUSHER

FIGURES FROM 370-BASIC COMPUTER PROGRAM, KBALGEO

MAX. PUSH ON CARRIAGE = 253.65 LB. (0° ELEV.)

TWO CHAINS - 126.83 LB. PER CHAIN.

SPROCKET RADIUS = 4.0815 IN.

MAX. CHAIN TRAVEL (75° GUN ELEV.) = 64.073 IN.

SPROCKET TORQUE = 253.65 x 4.0815 = 1035.27 [#]IN.

BOTH SPUR GEAR & BEVEL GEAR SETS HAVE
SAME SIZE GEARS - NO MECHANICAL ADVANTAGE

∴ TORQUE IN = TORQUE OUT.

FIND SPROCKET RPM. (MAX): $\frac{64.073}{2\pi \cdot 4.0815} = 2.47897$
RPM

THE HYD. MOTOR TORQUE = 2309 LB. IN. @ 1000 PSI, AND

THE RPM = 0-300 (5 RPS.). IF 1.5 SEC. IS ALLOWED

FOR THE CARRIAGE TRAVEL - $\frac{2.5}{1.5} = 1.667$ RPS. (100 RPM)

THE MOTOR IS ADEQUATE:

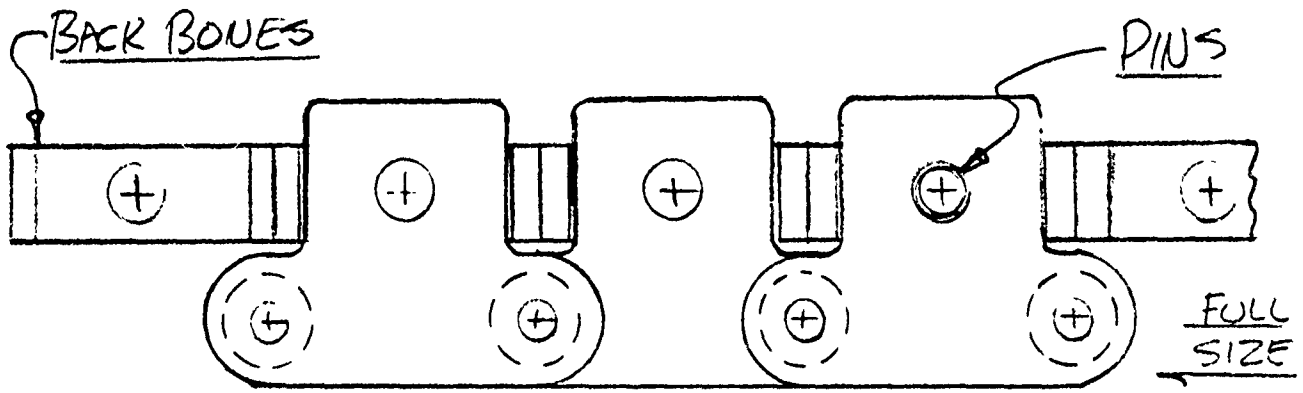
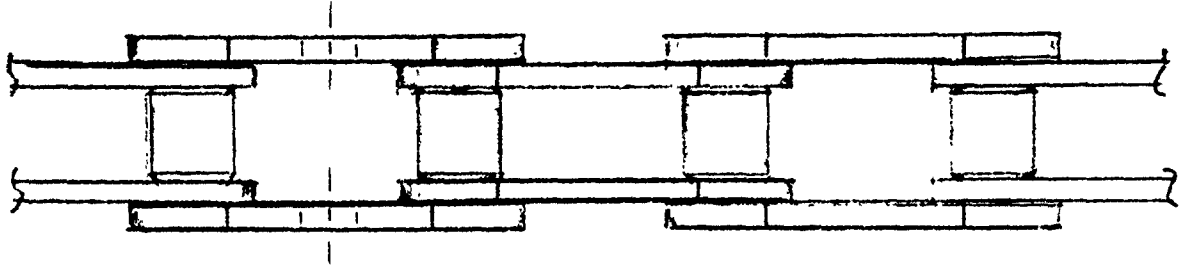
$$\frac{2309}{1035.27} = 2.23 \text{ F.S.}$$

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME K. BOWEE
DATE 11-6-79

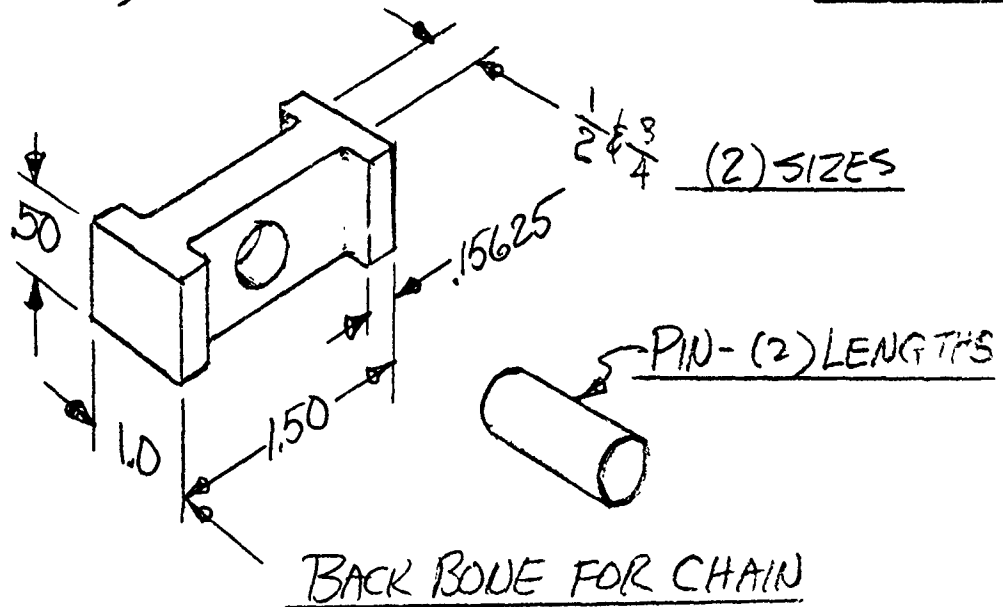
REFERENCE A-L
PAGE 16 OF

PUSH CHAINS



CHAIN - LINK BELT, DOUBLE PITCH, #RC1260 +M1
(ASA C2060)

ATTACHMENT

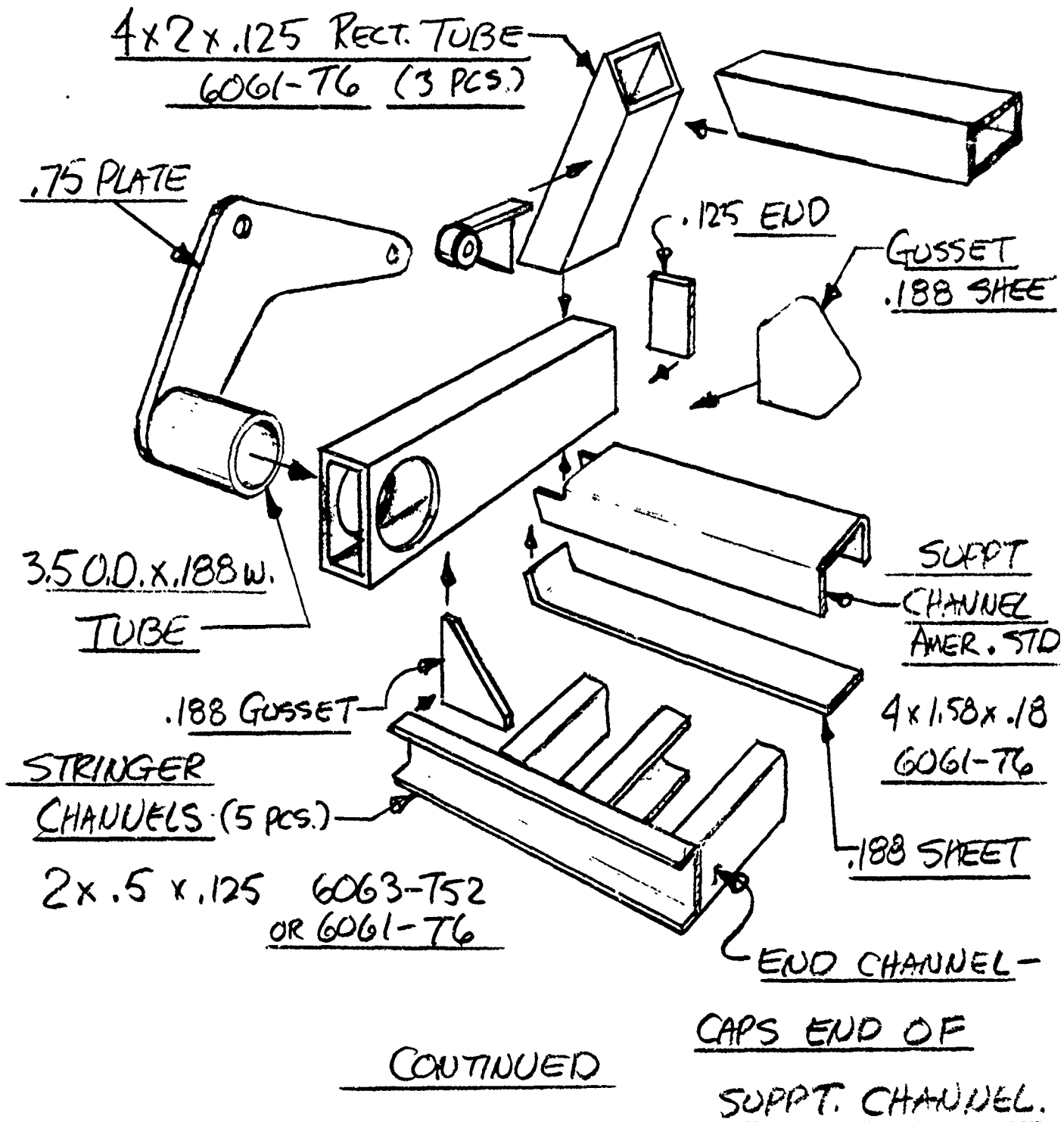


PACIFIC CAR AND FOUNDRY COMPANY
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NAME K. BOUZE
DATE 11-13-79

REFERENCE A-L CARRIAGE
PAGE 17 OF

CARRIAGE WELDMENT



PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME

K. Bover

REFERENCE

A-L. CARRIAGE

DATE

11-13-79

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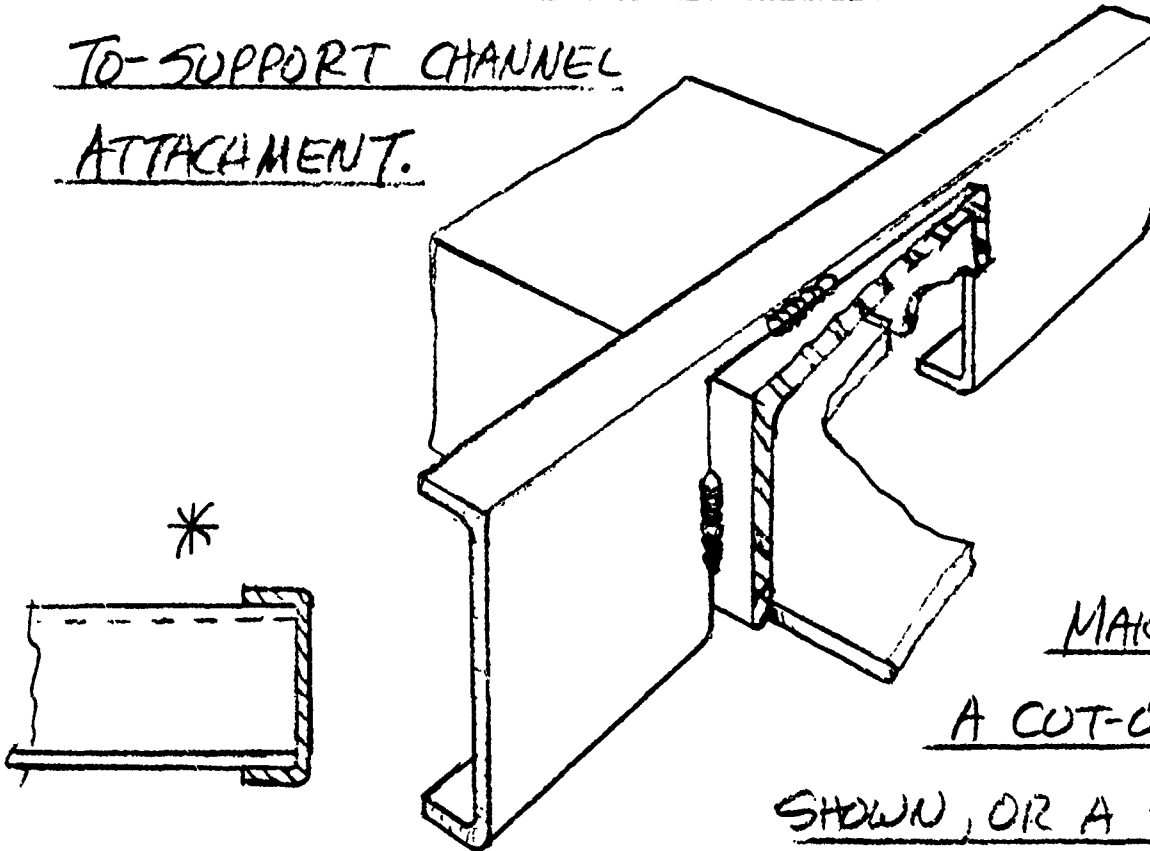
18

OF

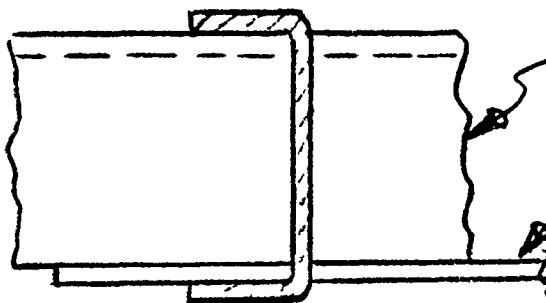
CARRIAGE WELDMENT

CONT.

TYPICAL STRINGER CHANNEL-
TO-SUPPORT CHANNEL
ATTACHMENT.



MAKE EITHER
A CUT-OUT AS
SHOWN, OR A SQ. HOLE
IF POSSIBLE, IN THE
STRINGER.*
CHANNELS.



CHANNEL

SHEET

*NOTE:

NO CUT-OUT IN THE
END STRINGER CHANNEL.

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME

K. B. BOE

REFERENCE

A-L. CARRIAGE

DATE

11-12-70

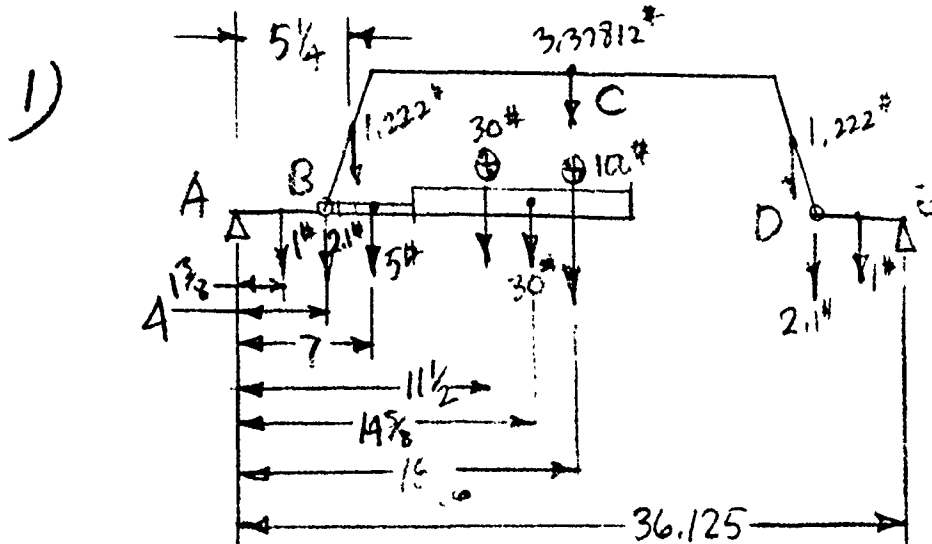
PAGE

19

OF

STRESS CALCS.

LOADS & MOMENTS



GENERAL

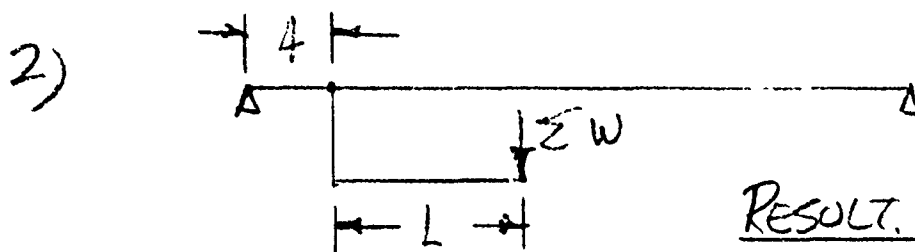
ESTIMATE

WT. DIAGRAM

DIST. B. W. S.

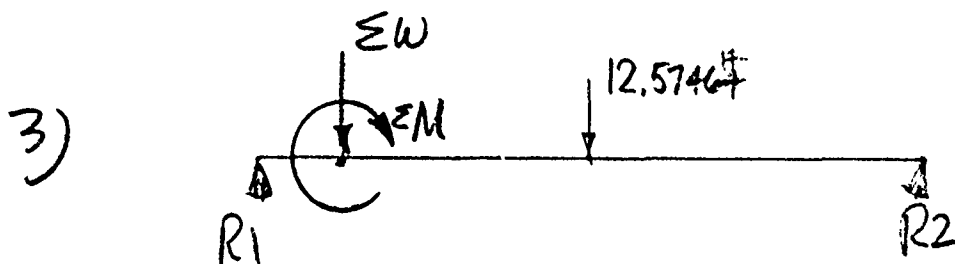
CALC. AS POINT

LOADS.



RESULT. OF LOADS ON

SUPP.T. CHANNEL.



$$\Sigma W = 100 + 30(2) + 5 = 165^{\#}$$

$$\Sigma M = 100(14 \frac{1}{16}) + 30(10 \frac{5}{8}) + 30(7.5) + 5(3) = 1965 \text{ IN. LB.}$$

$$L = 11.91'$$

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME H. B. Jett
DATE 11-12-79

REFERENCE A-L. Calkins
PAGE 20 OF

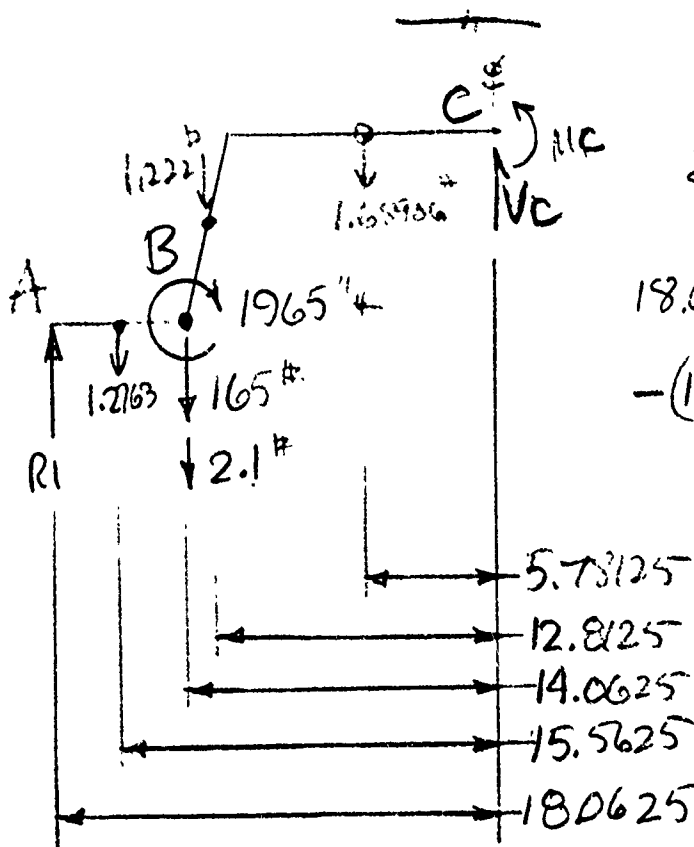
STRESS CALC.

LOADS & MOMS. CONT.

$$R_1 = -\frac{1965}{36.125} + \frac{165(32.125)}{36.125} + \frac{12.57464}{2} = \underline{98.62296}^{\#}$$

$$R_2 = \frac{1965}{36.125} + \frac{165(4)}{36.125} + \frac{12.57464}{2} = \underline{78.95168}^{\#}$$

4)



$$\sum M_C = 0;$$

$$18.0625 R_1 + 1965 - 1.2763(15.5625)$$

$$- (165 + 2.1) 14.0625 - 1.222(12.8125)$$

$$- 1.68906(5.78125) = M_C$$

$$M_C = \underline{1351.2493}^{\#}$$

$$\sum V = 0$$

$$V_C = 171.28736 - R_1$$

$$V_C = \underline{+72.94072}^{\#}$$

CHECK: $M_C = R_2(18.0625) - 1.2763(15.5625)$

$$- 2.1(14.0625) - 1.222(12.8125) - 1.68906(5.78125) = 1351.2493$$

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME

K. BOVEE

REFERENCE

A-L. CARRIAGE

DATE

11-12-79

PAGE

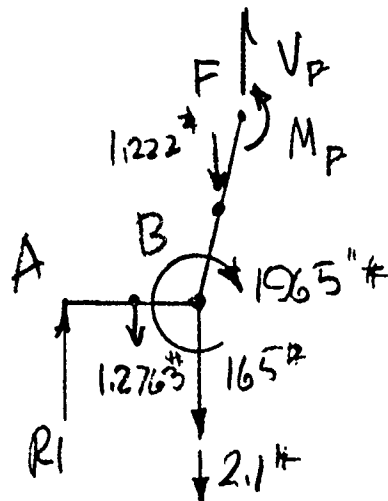
21

OF

STRESS CALCS.

LOADS & MOM. CONT.

5)



$$\sum M_F = 0$$

$$R_1(6.5) - 1.2763(4)$$

$$- 167.1(2.5) - 1222(1.25)$$

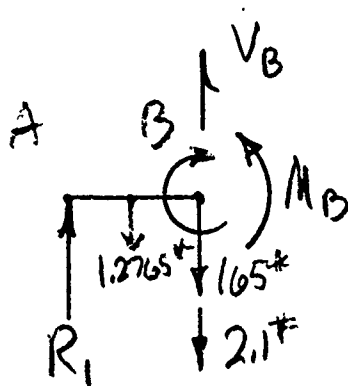
$$+ 1965 = M_F = \underline{2181.666 \text{ inch-kips}}$$

$$\sum V_F = 0$$

$$(1.2763 + 165 + 2.1 + 1.222) - R_1 = V_F$$

$$V_F = \underline{70.97534 \text{ kips}}$$

6)



$$\sum M_B = 0$$

$$R_1(4) + 1965 - 12765(2.625)$$

$$= M_B = \underline{2356.14103 \text{ inch-kips}}$$

$$\sum V_B = 0$$

$$165 + 2.1 + 1.2765 - R_1 = V_B$$

$$V_B = \underline{69.75354 \text{ kips}}$$

(CONT.)

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME

K. BOVEE

REFERENCE

A-L. CARRIAGE

DATE

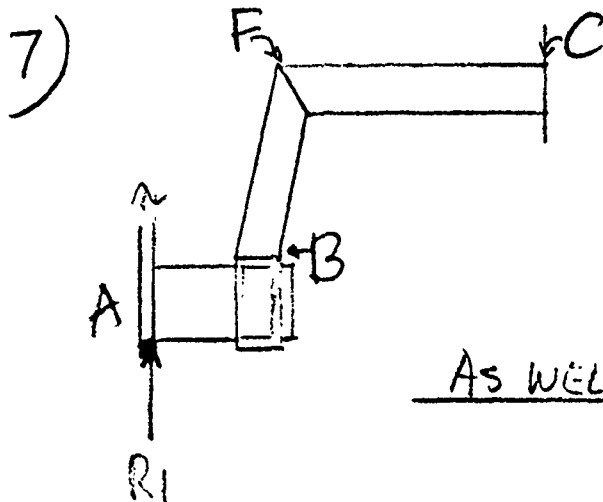
11-12-79

PAGE

22

OF

STRESS CALCS.



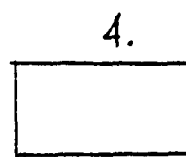
WELD LINE ANALYSIS

TUBING - AL AL - 6061-T6

$F_{Ty} = 35,000 \text{ PSI}$

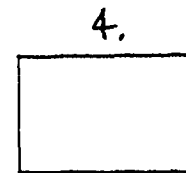
AS WELDED $F_{Ty} = 9000 \text{ PSI}$ (NO H.T.)

a) CROSS SECT. @ C



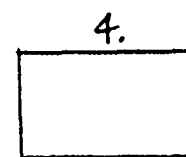
2. $M = 1351.25 \text{ in}^4$
 $V = 72.941 \text{ in}^3$

b) CROSS SECT. @ F



2.5 $M = 2181.67 \text{ in}^4$
 $V = 70.93 \text{ in}^3$

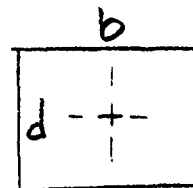
c) CROSS SECT @ B



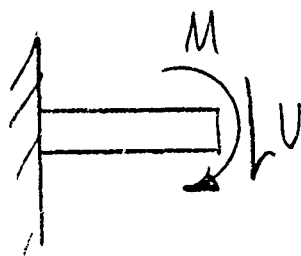
2.0706 $M = 2356.141 \text{ in}^4$
 $V = 69.754 \text{ in}^3$

FOR RECT. WELD PATTERN ~

$S_w = bd + \frac{d^2}{3}$; $J_w = \frac{(b+d)^3}{6}$; $A_w = 2(b+d)$



REF. DESIGN OF WELDMENTS - BLODGETT



FOR BENDING

$f_b = \frac{M}{S_w}$

$f_s = \frac{V}{A_w}$

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME KBOEE
DATE 11-12-79

REFERENCE A-L, CARRIAGE
PAGE 23 OF

STRESS CALCS

$$7a) \quad f_b = \frac{1351.25}{9.34} = 144.67345 ; S_w = 4(2) + \frac{2^2}{3} = 9.34$$

$$f_s = \frac{72.941}{12} = 6.078417 ; A_w = 2(4+2) = 12$$

$$f_r = \sqrt{f_b^2 + f_s^2} = 144.8 \text{ #/in. WELD LINE}$$

STRENGTH REQ'D

$$7b) \quad f_b = \frac{2181.67}{12.0833} = 180.5525 ; S_w = 12.08333$$

$$f_s = \frac{70.98}{13} = 5.46 ; A_w = 13$$

$$f_r = 180.635 \text{ #/in. REQ'D}$$

$$7c) \quad f_b = \frac{2356.141}{9.71153} = 242.61275 ; S_w = 9.71153$$

$$f_s = \frac{69.754}{12.1412} = 5.74523 ; A_w = 12.1412$$

$$f_r = 242.681 \text{ #/in. REQ'D}$$

(CONT.)
A-23

NAME K. BOVEE
DATE 11-12-79

REFERENCE A-L, CARRIAGE
PAGE 24 OF

STRESS CALCS

WELD LINE STRENGTH REQUIS.

FOR 6061-T6, AS WELDED, NO HEAT TREAT,
AND USING A FATIGUE FACTOR FOR 10^6 CYCLES —

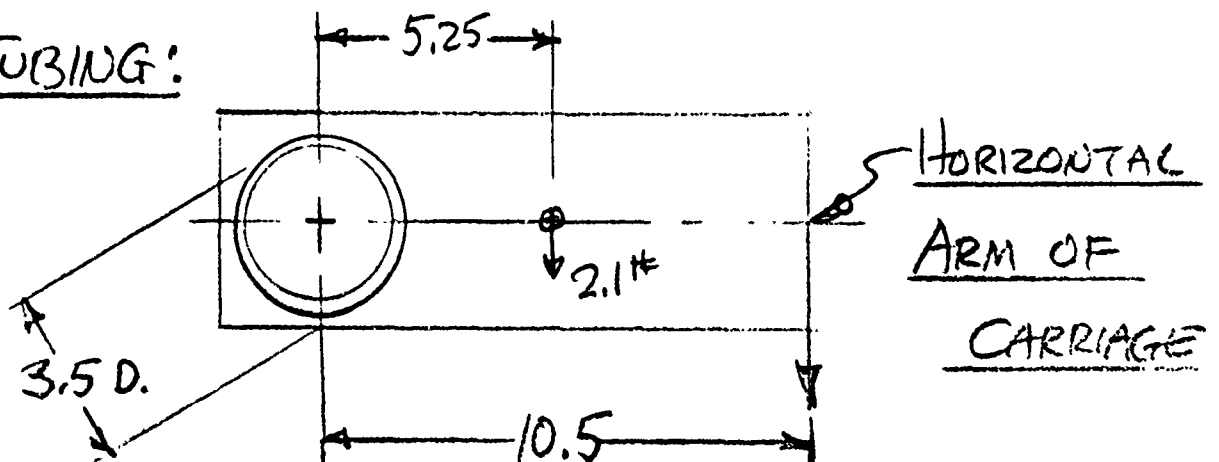
$$f = .707 \overset{(\Delta)}{(\text{WELD LEG})} \overset{(F_y)}{9000} \overset{(F.F.)}{(.61)} / \underset{(F.S.)}{2}$$

USING $f_r = 243 \text{ #/in}^2$ (CALC 7C.)

$$(\text{WELD LEG}) = \frac{243(2)}{.707(9000).61} = .125211584"$$

\therefore A WELD \geq A FILLET WELD OF $\frac{1}{8}$ LEG IS O.K.

8) TUBING:



$$J_w = \frac{\pi D^3}{4} = 33.674'; \quad f_t = \frac{1774.08(1.75)}{33.674} = 92.197$$

$$A_w = \pi D = 10.9956'; \quad f_s = \frac{17001}{10.9956} = 15.46164 \text{ #/in}$$

(CONT.) A-24

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME K. BOVEE
DATE 11-12-79

REFERENCE A-L, CARRIAGE
PAGE 25 OF

STRESS CALCS.

(CONT.)

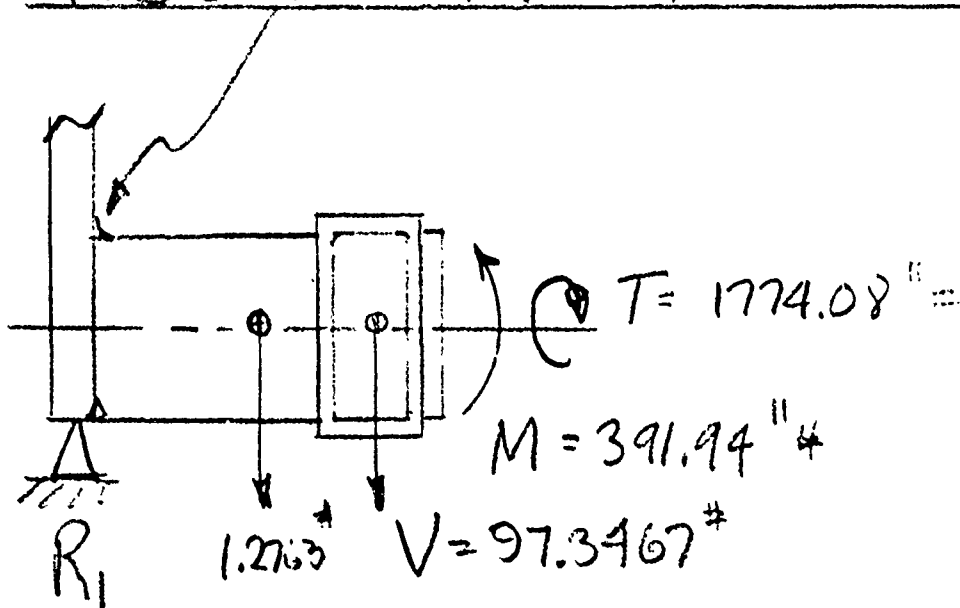
8) CONT.

$$P_r = \sqrt{P_t^2 + P_s^2} = 93.4845 \text{ #/in.}$$

HAVE (2) 1/8" FILLET WELDS @ 242.6 #/in.

A PIECE, SO $\frac{2(242.6)}{93.5} = \text{F.S. } 5.19 \text{ EXTRA}$

9) TUBE WELD AT L.H. SUPPORT:



$$J_w = 33.674$$

$$A_w = 10.9956$$

$$S_w = \frac{\pi D^2}{4} = 9.62113$$

$$P_t = \frac{1774.08(1.75)}{33.674} = 92.19695$$

$$P_s = \frac{98.6226}{10.9956} = 8.96931$$

$$P_b = \frac{391.9394}{9.62113} = 40.73736$$

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME V. B. E. E.
DATE 11-12-77

REFERENCE J. L. CARR 4-15
PAGE 26 OF

STRESS CALCS.

(CONT.)

9/CONT.

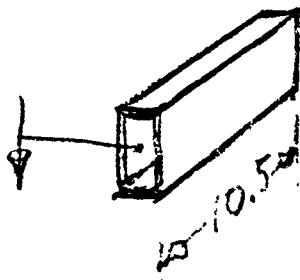
$$\sqrt{(f_c + f_t)^2 + f_b^2} = 109.06 \text{ #/in.}$$

WELD STRESS REQ.

$$(\text{LEG LGTH.}) = \frac{109.06(2)}{.707(9000).61} = .0562 \text{ IN.}$$

ALLOW 6061-T6 AS WELDED, 10° CIRCLES.
THE TUBE WALL IS 7/16" & THE SUPPORT
ARM IS 7/8" DIA. 6061 T6, HENCE WELD
WELD 1/16" & UP IS O.K.

10) CALC. TORS. DEFL. OF HORIZ. ARM.



$$T_u = R_1(4) + 1965 = 2359.412 \text{ #}$$

$$R = 2.2952 \quad \text{F.S.} = 15.4$$

$$\tau = \frac{T}{2t(b-t)(d-t)} = 1298.99355 \text{ PSI}$$

$$R = \frac{2t(b-t)^2(d-t)^2}{b+d-2t}$$

$$\Theta = \frac{TL}{GR} = \frac{2359.5(10.5)}{375000(2.2952)} = .0023784419 \text{ rad}$$

$$= .165^\circ$$

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME

K. B. E. E.

DATE

11-12-79

REFERENCE

A-L. CARRIAGE

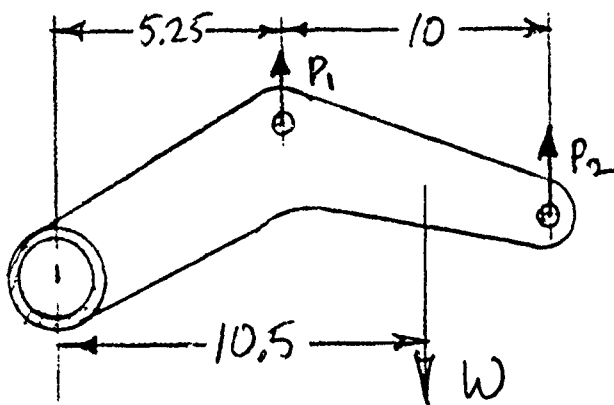
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OF

SUPPORT ARMS

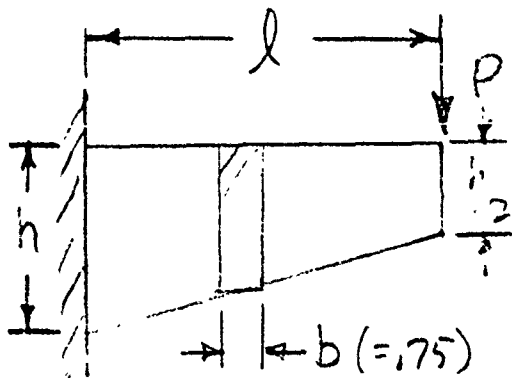
11)



$$P_1 = .475 W = 87.875^{\#}$$

$$P_2 = .525 W = 97.125^{\#}$$

$$W = 185^{\#} \text{ (ASSUMED)} \\ (2 + F.S.)$$



$$h = \sqrt{\frac{6 P l}{b S_a}} \quad \text{MECH. ENG. HANDBOOK}$$

$$h_{1a} = \sqrt{\frac{6 P_2 (10)}{.75 (20,000)}} = .6233^{\prime}$$

$$h_2 = \sqrt{\frac{6 P_1 (5.25)}{.75 (20,000)}} = .4296^{\prime} \quad ; \quad h_3 = \sqrt{\frac{6 (P_1 + P_2) (5.25)}{.75 (20,000)}} = .3885^{\prime}$$

$$h_{1b} = \sqrt{\frac{6 P_2 (15.25)}{.75 (20,000)}} = .769716^{\prime} \\ \hline 1.1993^{\prime}$$

$$\text{ACTUAL "h" IS} = 3.57^{\prime} \quad \therefore \frac{3.5}{1.2} = 2.9 \text{ F.S. (LIMIT)}$$

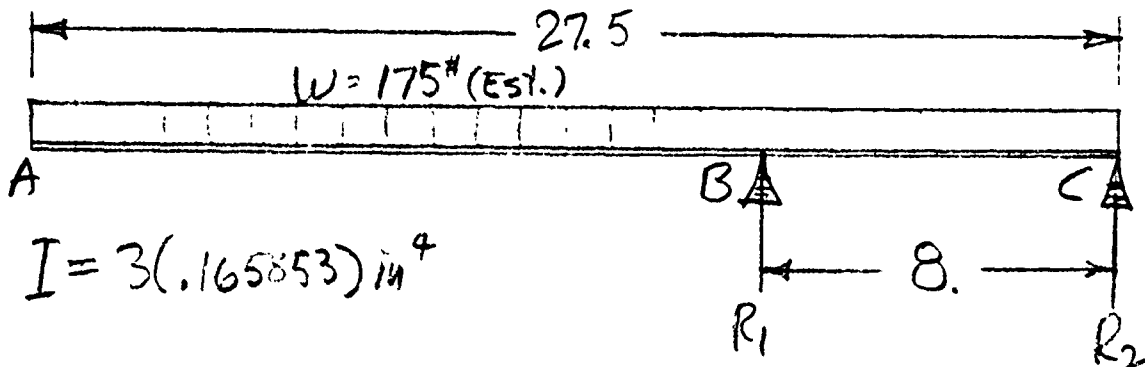
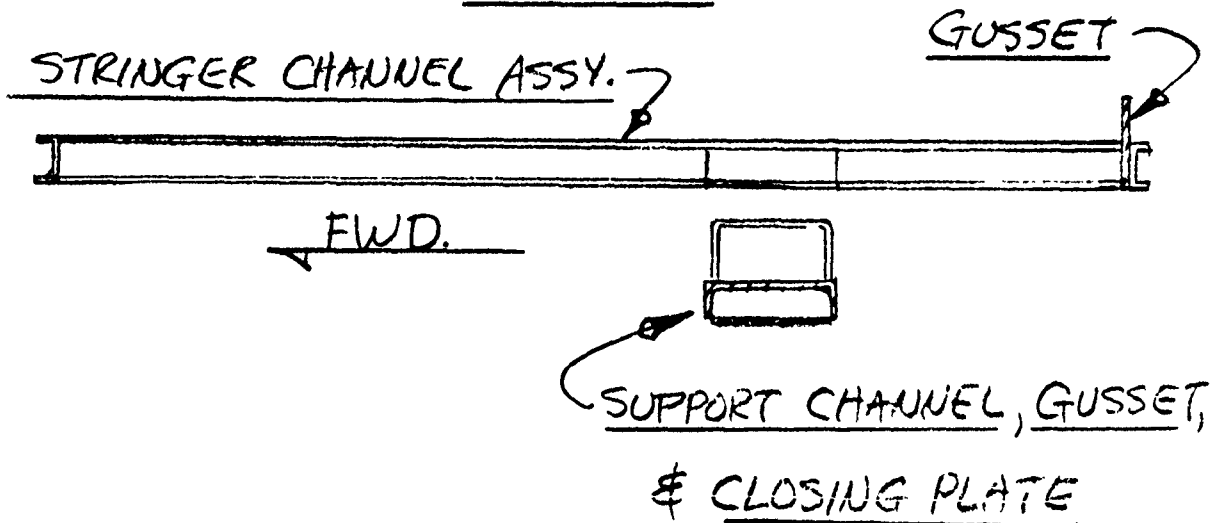
PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME K. BOUZE
DATE 11-13-79

REFERENCE A-L CARRIAGE
PAGE 28 OF

CARRIAGE WELDMENT

CONT.



$$I = 3(.165853) \text{ in}^4$$

$$M_{\text{max. @ B}} = -1209.89 \text{ " \#}$$

$$F_{t_{\text{max}}} @ B = \frac{M c}{I} = 2431.64 \text{ psi.}$$

$$F.S. = \frac{11,000 \text{ psi.}^*}{2431.64(2)} = 2.26 \text{ (ADDITIONAL.)}$$

* $F_{ty} = 11,000 \text{ psi}$ FOR 6061-T6 AS WELDED.

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME

K. BOVEE

REFERENCE

A-L. CARRIAGE

DATE

11-14-79

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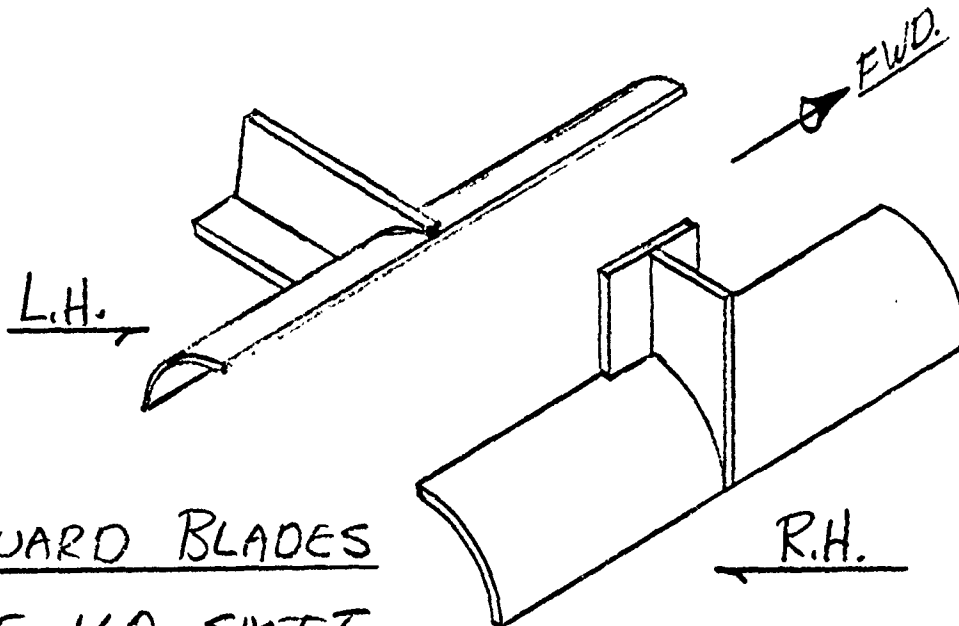
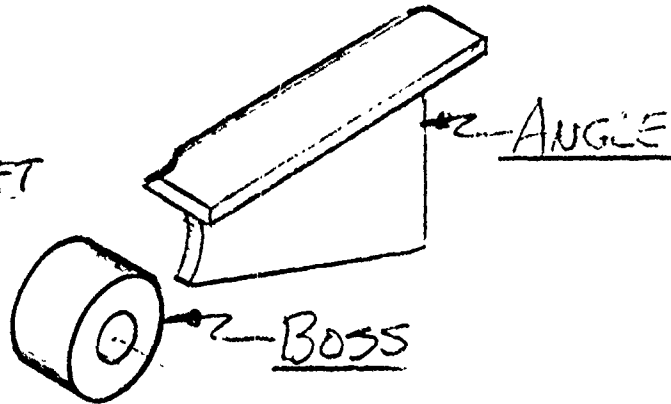
OF

CARRIAGE WELDMENT

MISCELLANEOUS PARTS:

CHAIN LUG.

L. & R. H. BRACKET



GUARD BLADES

.125-.160 SHEET

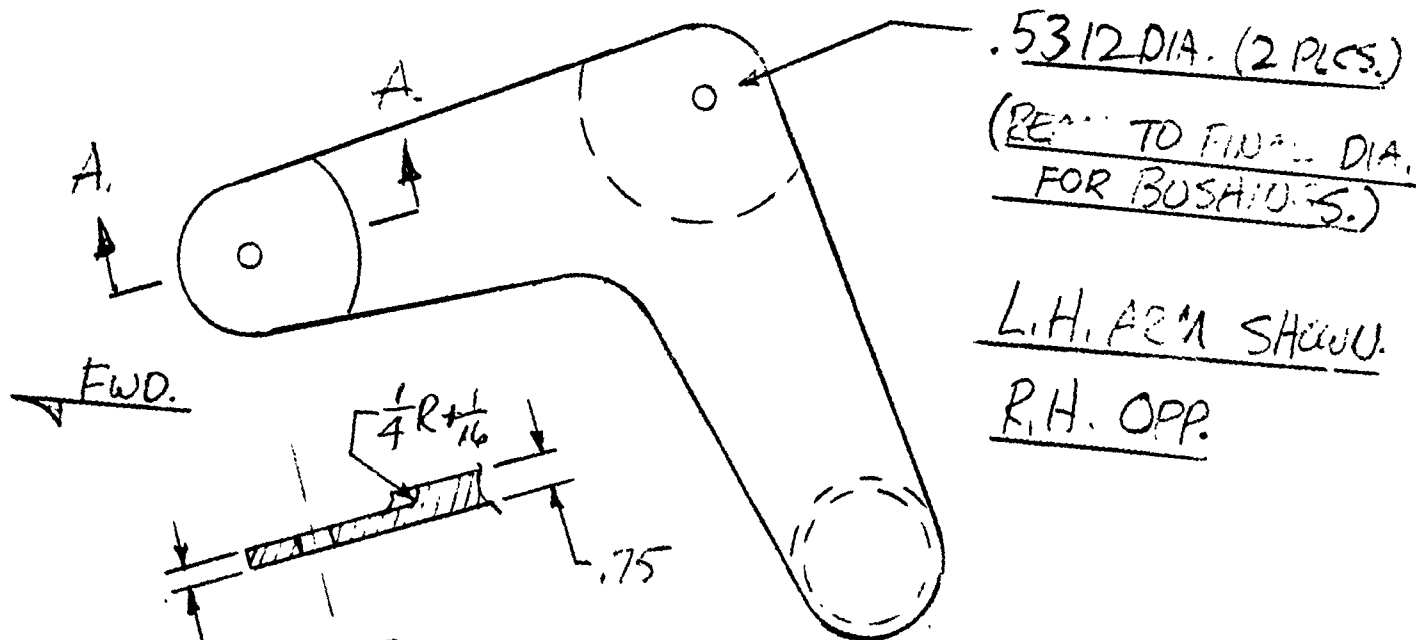
ALAL 6061-T6.

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME K. B. E. E.
DATE 11-14-79

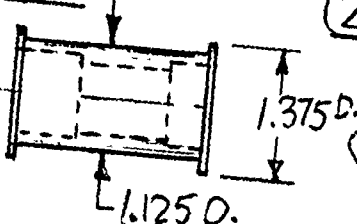
REFERENCE A-L. CARRIAGE
PAGE 30 OF

SUPPORT ARMS

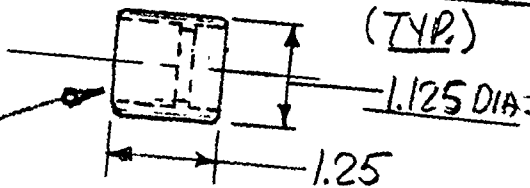


ROLLER, UPPER

SECT. A-A
DIMS. TYP.
(2 PLCS.)



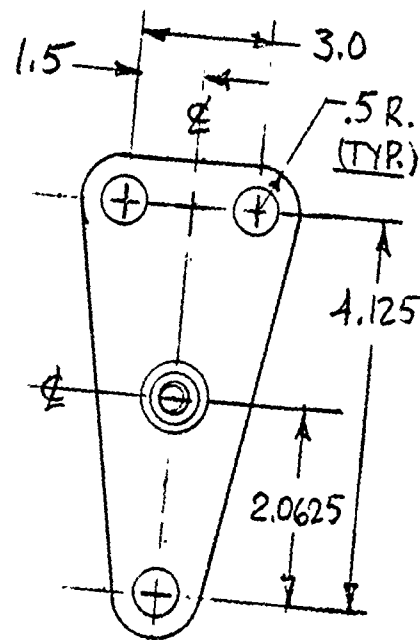
NUT, WASHER, SPACER, BRG., SPACER, BRG.
(TYP.)



ROLLER, LOWER

AXLE & STUDS

* FLANGED BUSHINGS (ON ASSY.)

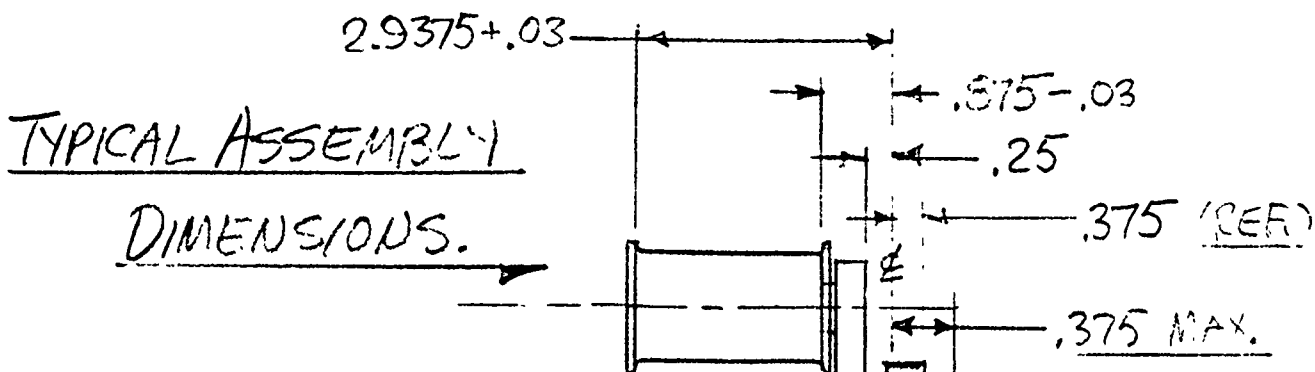


PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME K. BOVEE
DATE 11-15-79

REFERENCE A-L, CARRIAGE
PAGE 31 OF

SUPPORT ARMS



SUGGESTED BEARINGS -

FAFNIR 5-INCH SERIES

#53K PLAIN .2188 W.

"53KD (1) SHIELD .2812 W.

"53PP (2) SEALS .2812 W.

O.D. - .875" D.

BORE - .375" D.

.016 FILLET RAD.

LOAD RPM

575#	33 1/3
505	50
400	100
317	200
277	300
234	500

ESTIMATED LOAD PER BEARING = $\frac{320\#}{16} = 20\#$ PER BRG.

NAME

KBUSEE

DATE

11-8-79

REFERENCE

A-L TRAYS

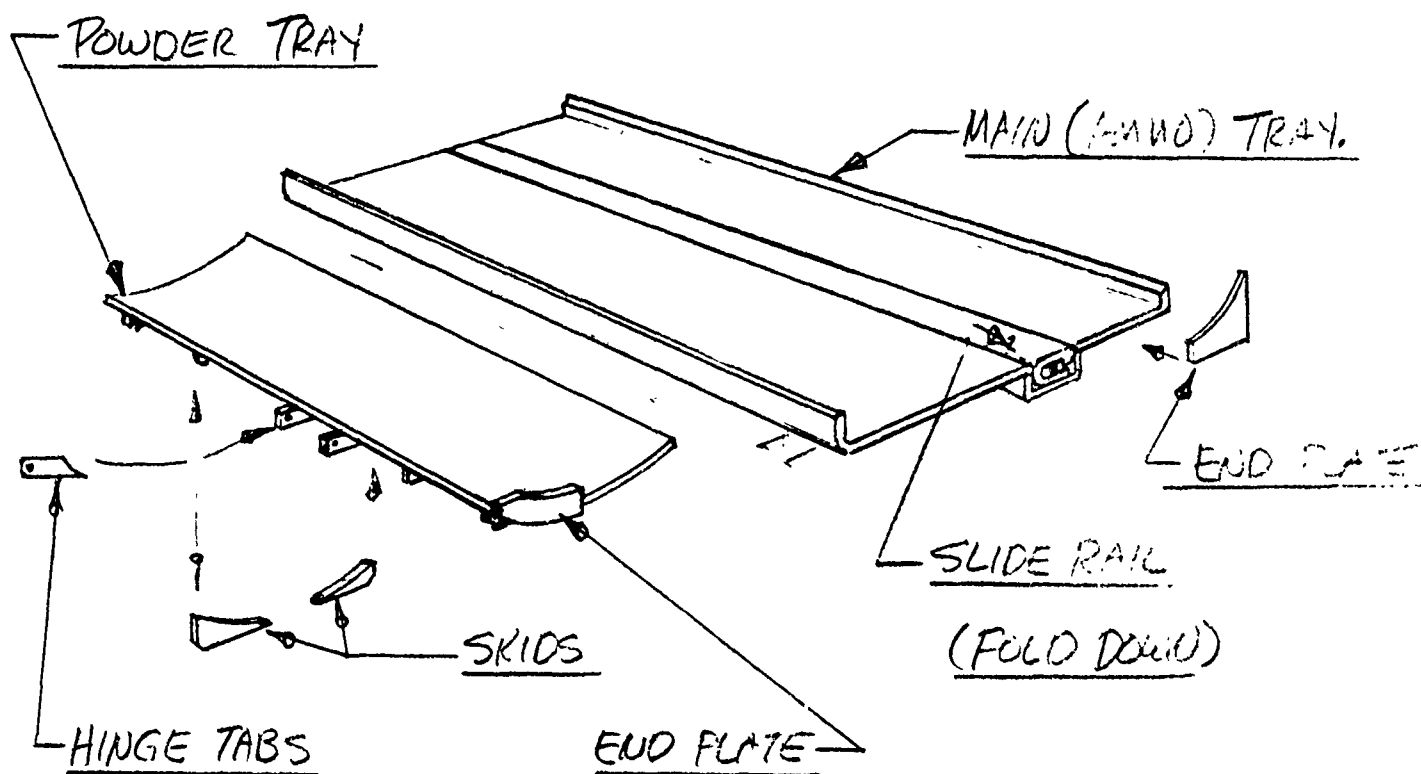
PAGE

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OF

TRAYS

THE POWDER TRAY IS MOVABLE BUT MAIN TRAY
SITS 'FIXED' ON THE CARRIAGE FRAME. IT CAN
MOVE UP BUT NOT SIDEWAYS.



THE END PLATES HOLD THE CANNISTER &
ROUND ON THE TRAYS WHEN THE CARRIAGE
ROTATES OFF HORIZONTAL. THE TRAYS ARE POS-
ITIONED A LITTLE LOWER THAN THE DELIVERY
SHELF LIP SO AS TO MISS HITTING THE END PLATE
DURING LOADING.

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

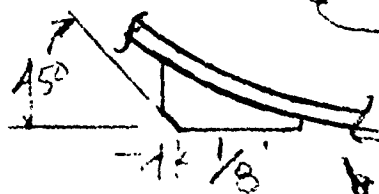
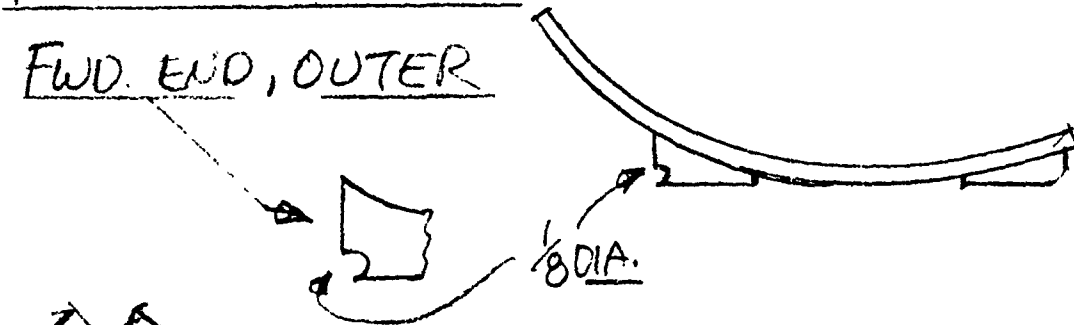
NAME B. J. E.
DATE 12-21-79

REFERENCE A-L. TRAYS
PAGE 33 OF

TRAYS

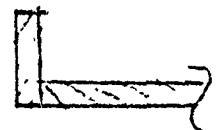
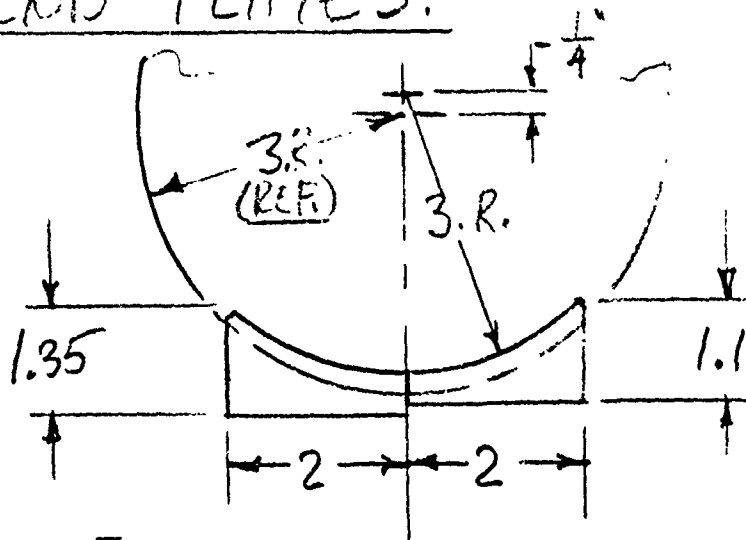
POWDER TRAY SKIDS:

FWD. END, OUTER

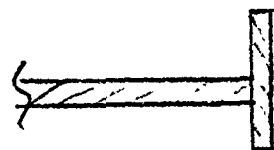


ALL OTHERS

END PLATES:



MAIN TRAY
X-SECT.



POWDER
TRAY
X SECT.



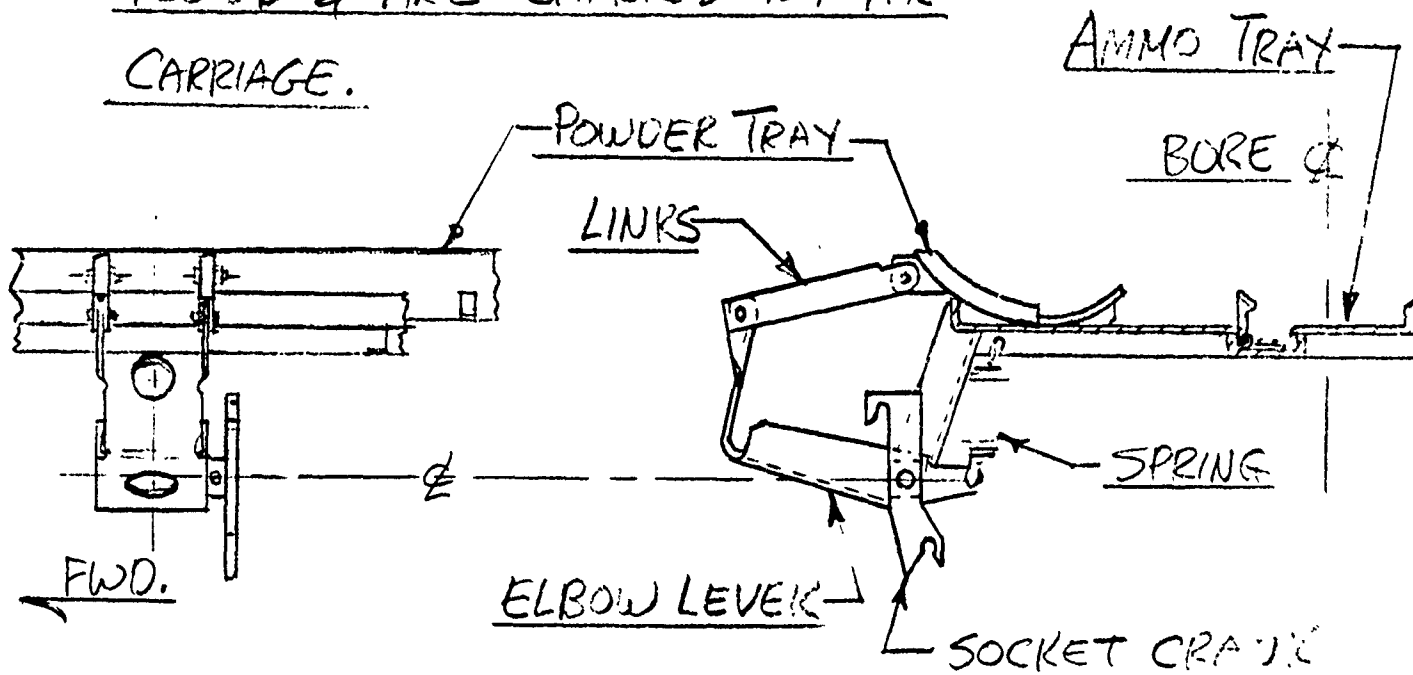
TYP. END. PLATE

NAME K. BOZZE
DATE 11-7-79

REFERENCE A-L TRAYS
PAGE 34 OF

TRAYS

THE TRAYS HOLD THE POWDER CANISTER & THE
ROUND & ARE CARRIED BY THE
CARRIAGE.



SPRING

$$d = .09", D = .72", N = 16.3$$

MUSIC WIRE, FULL

LOOP ENDS, CLOSED

LENGTH ~ 3", EXT. ~ 4.75"

INITIAL TENS. = 5#, EXT.

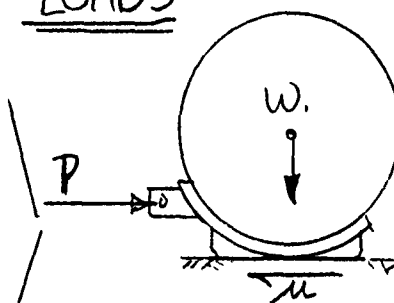
LOAD = 32#, $\frac{D}{d} = 8$

$$F_{Tens.} = 95,642 \text{ PSI.}$$

$$F_{Tens.} = 134,379 \text{ psi. (hook).}$$

PCF-RN-1284

LOADS



USE $W = 50\#$

$\mu_1 = .78$ (stk. on stk.)

$\mu_2 = .57$ (sliding)

Loaded $\left\{ \begin{array}{l} \rightarrow P_1 = \mu_1 N = .78(50) = 39\# \\ \rightarrow P_2 = \mu_2 N = 28.5\# \end{array} \right.$

Empty $\left\{ \begin{array}{l} \rightarrow P_3 = \mu_2 N_1 = .57(6.5\#) = 3.71\# \end{array} \right.$

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME

K. BOZZE

DATE

11-8-79

REFERENCE

A-L. TRAYS

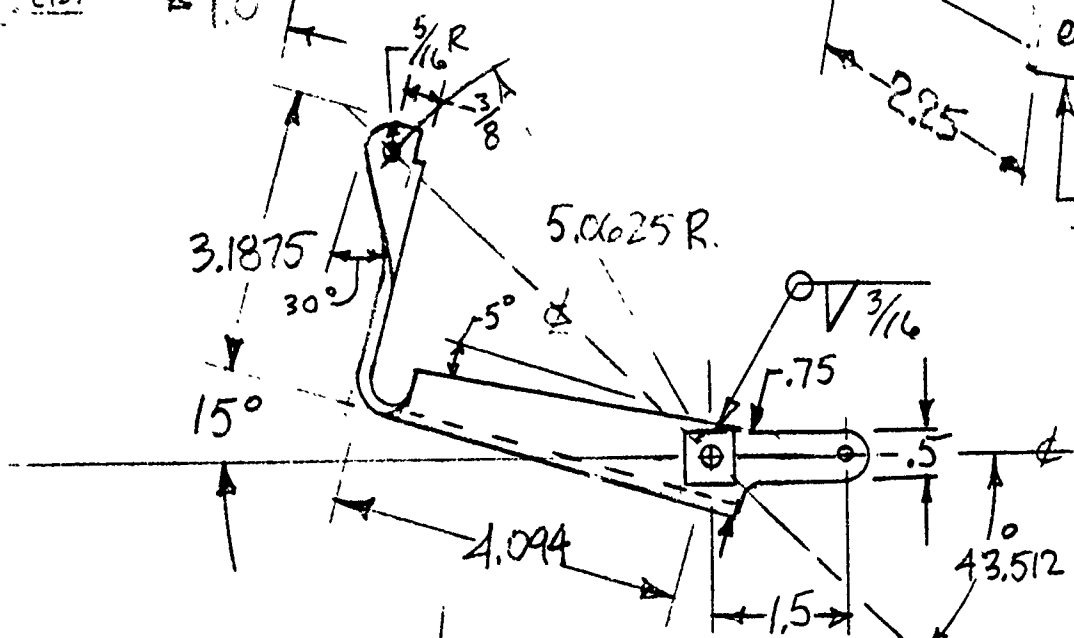
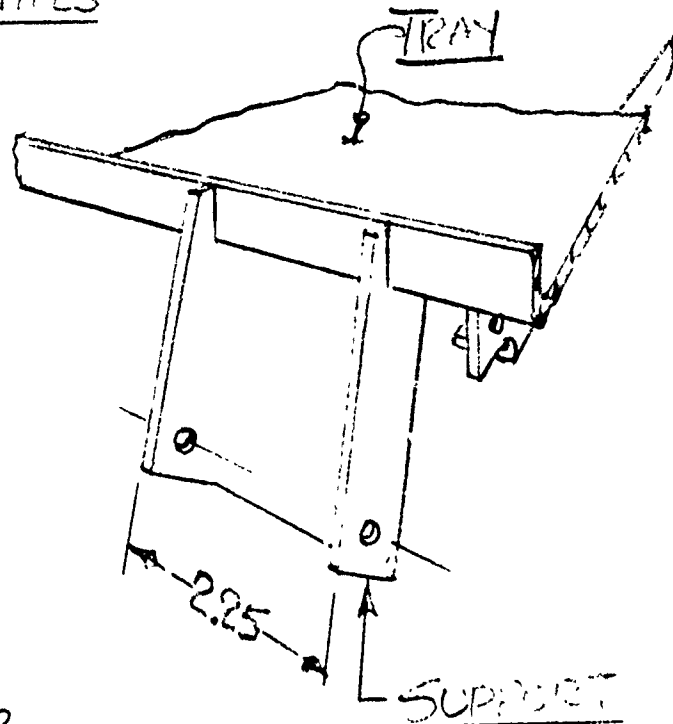
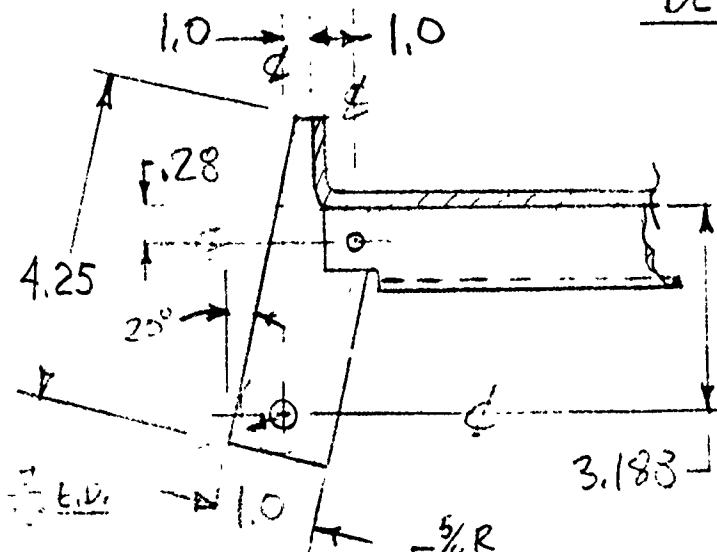
PAGE

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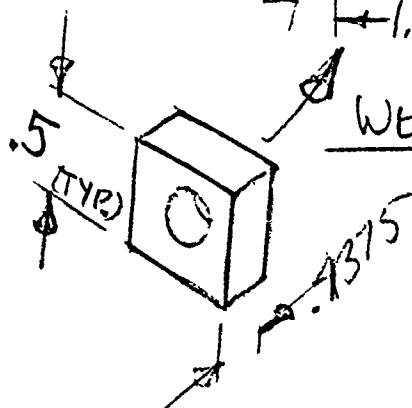
OF

MAIN TRAY

DETAILS



WELD TO LEVER



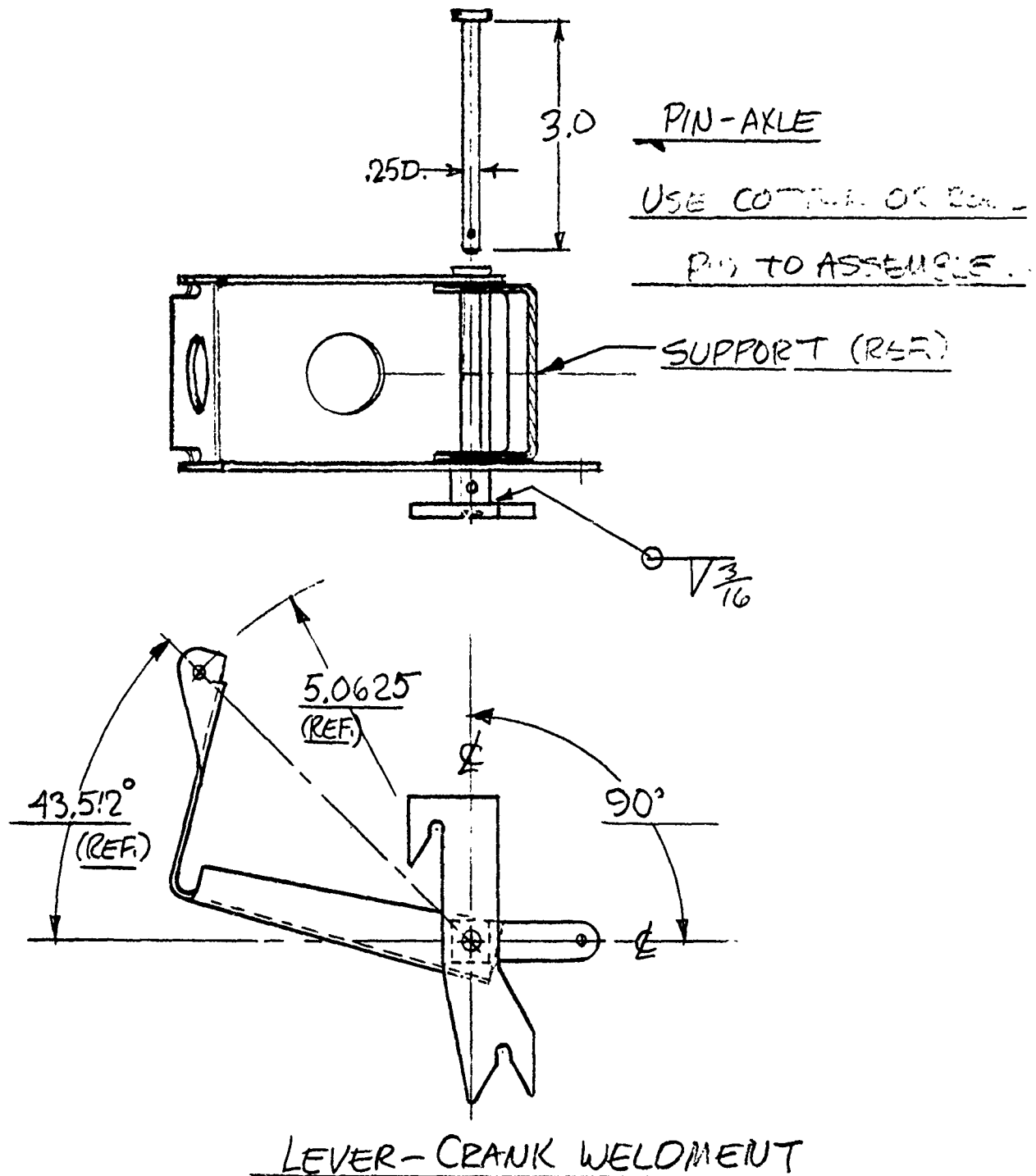
ELBOW LEVER

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME K. Boulee
DATE 11-9-79

REFERENCE A-L TRAYS
PAGE 36 OF

ELBOW LEVER ASSY.

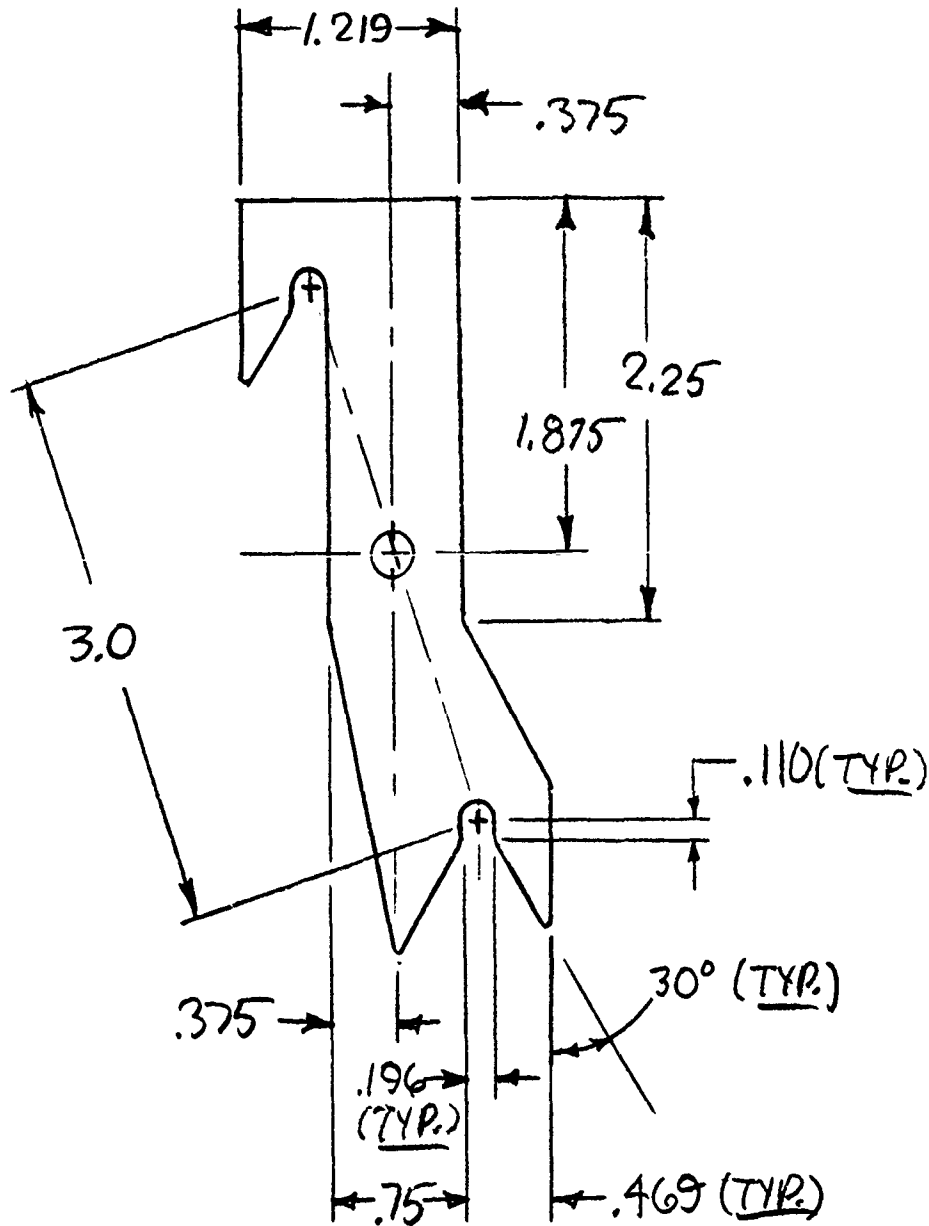


PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME 14 Bore
DATE 11-9-79

REFERENCE A-L TRAYS
PAGE 37 OF

SOCKET CRANK



MAKE FROM $\frac{3}{16}$ PLATE - 1025 STL. OR EQUIV.

BREAK ALL CORNERS & EDGES.

CRANK WELDS TO BLOCK ON ELBOW LEVER

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME

KBSEE

DATE

11-8-79

REFERENCE

A-L TRAYS

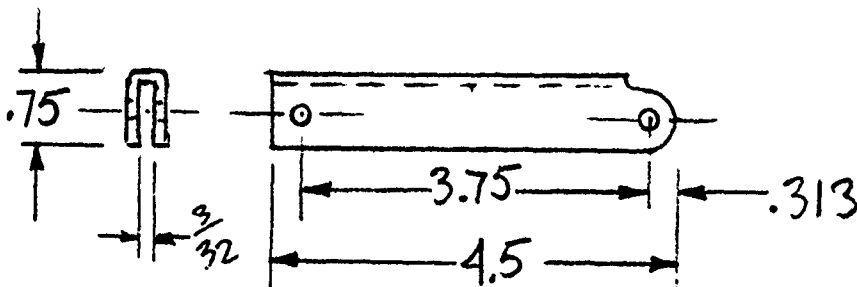
PAGE

39

OF

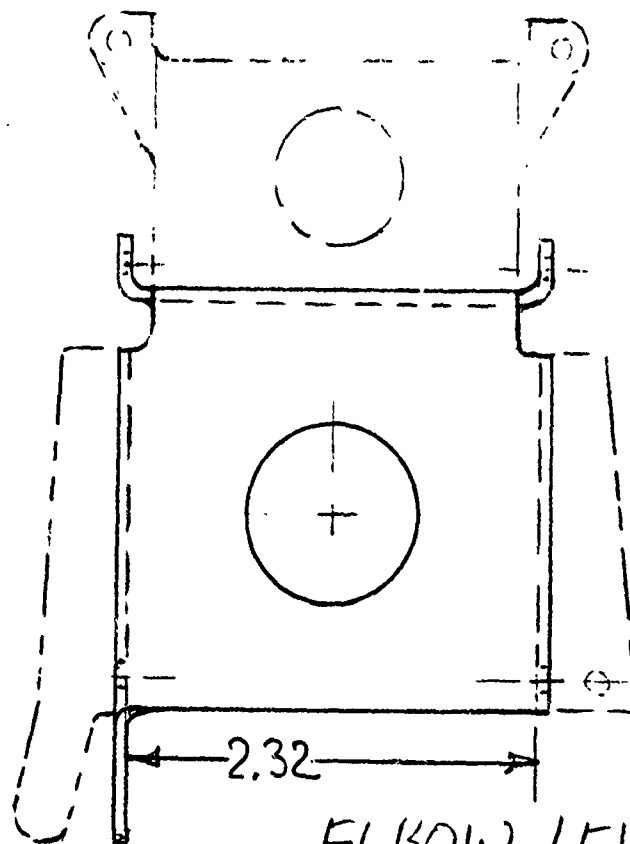
MAIN TRAY

DETAILS



LINK

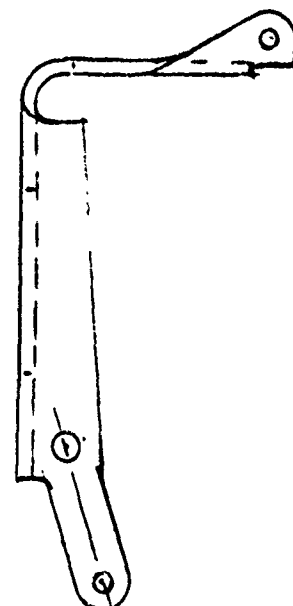
USE 16 GAGE STL.



ELBOW LEVER

(WITH FLAT PATTERN)

USE 1IN. DIA. LIGHTENING



11/11/80

(NO SCALE)

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME

K. BOWEE

DATE

9-7-79

REFERENCE

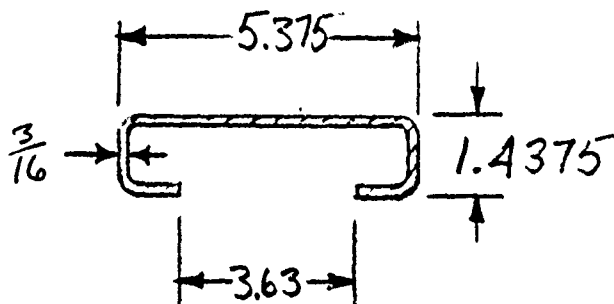
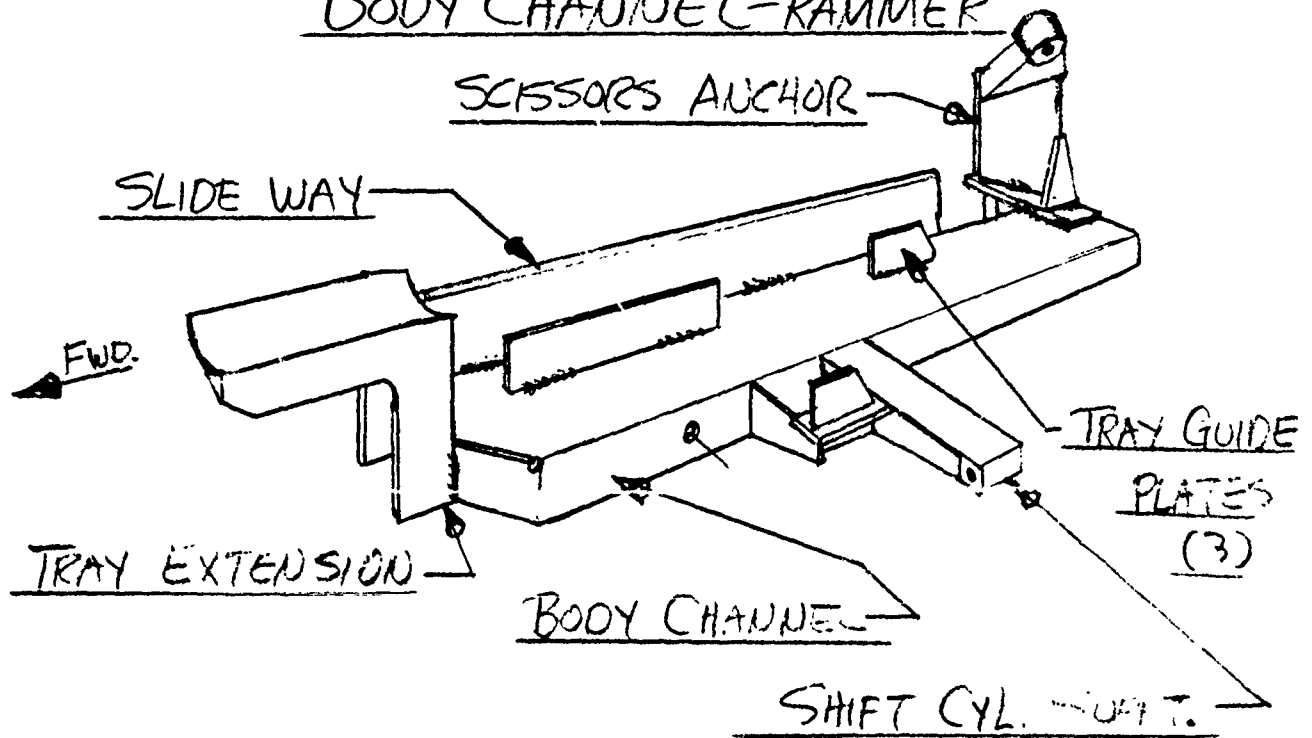
A-L RAMMER

PAGE

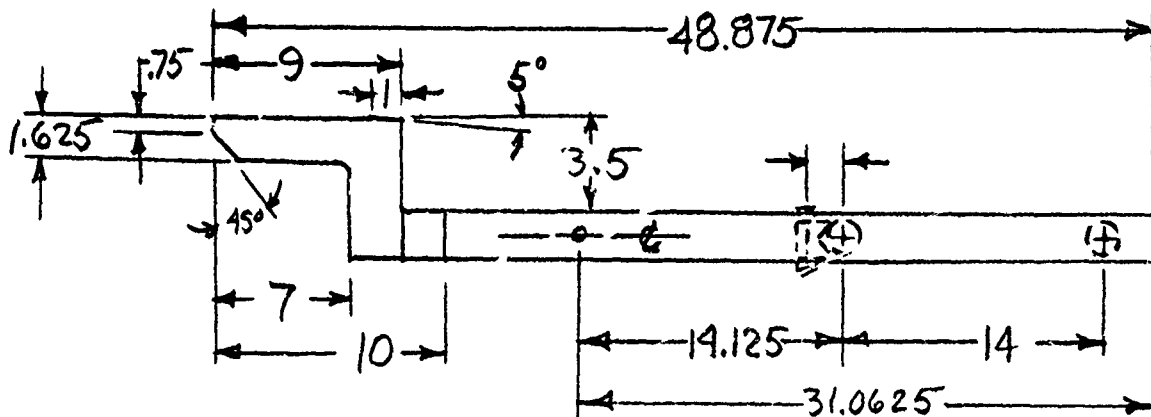
39

OF

BODY CHANNEL-RAMMER



CHANNEL CROSS-SECT.



GEN'L.
DIMS.

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME

K. BOZZE

DATE

9-10-79

REFERENCE

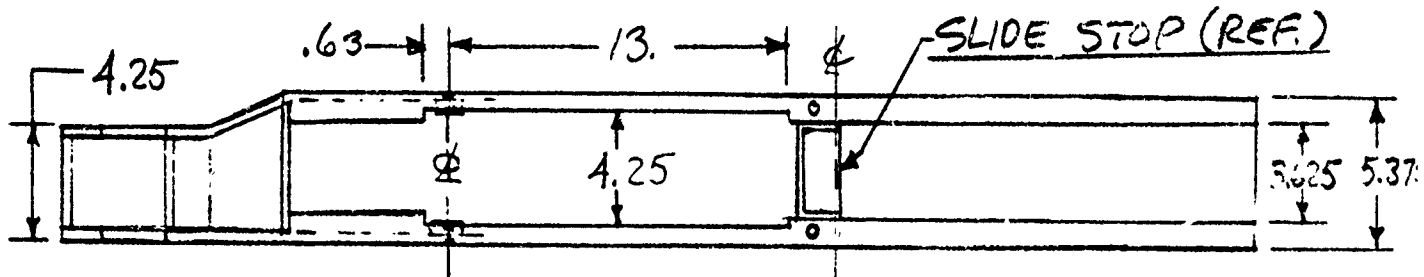
A-L RAMMER

PAGE

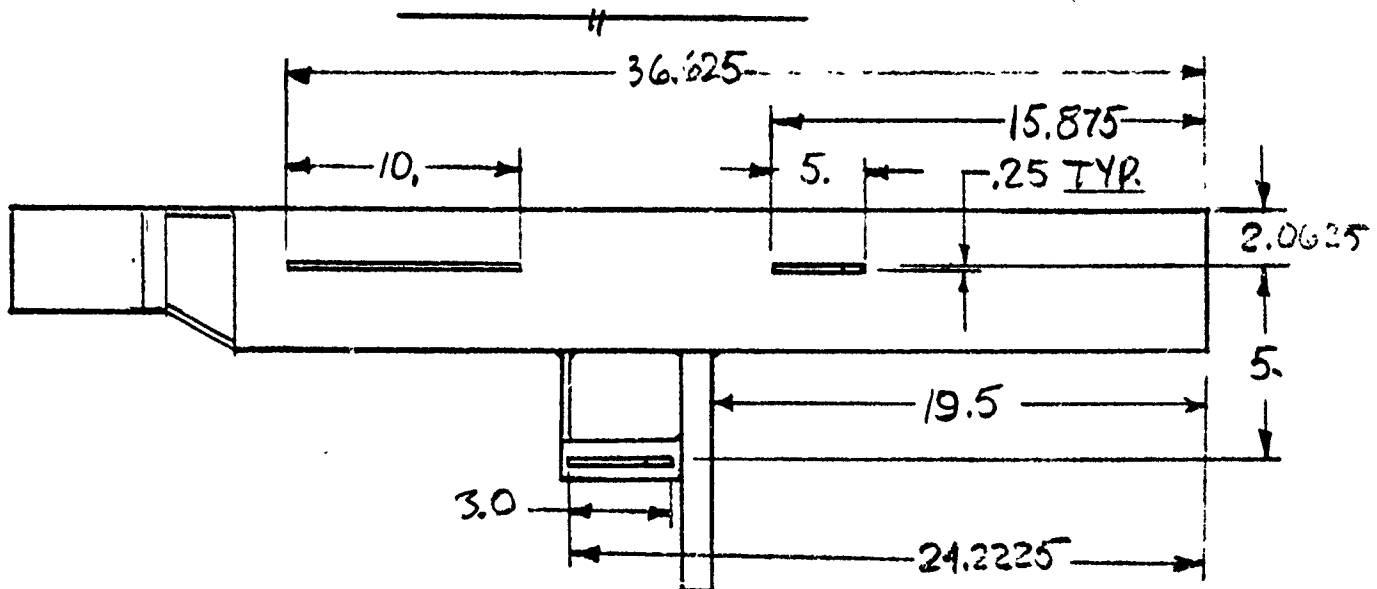
40

OF

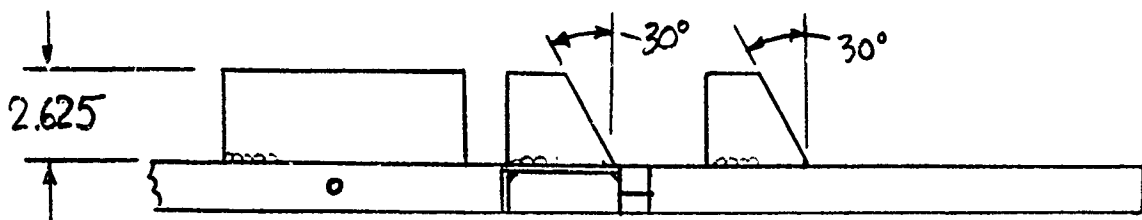
BODY CHANNEL - RAMMER



VIEW OF CHANNEL UNDERSIDE



VIEW OF CHANNEL TOP SIDE



TRAY GUIDE PLATES.

& SUPPT.

NAME
DATE 9-13-70

REFERENCE A-L, Enmap
PAGE 41 OF

[illegible]

CYL. IS TOM THUMB SERIES AV, P (MP3); $\frac{3}{4}$ BORE,
 $\frac{1}{4}$ ROD DIA., (2) CUSHIONS, 1.50 STROKE; 6.6875" LENGTH
 ROD CLEVIS \neq TO MOUNT \neq , RETRACTED; 500 PSI HYDR. PRESS.
 MAX. PUSH = 220[#]

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME

K. B. JEE

DATE

11-30-79

REFERENCE

A-L RANIER

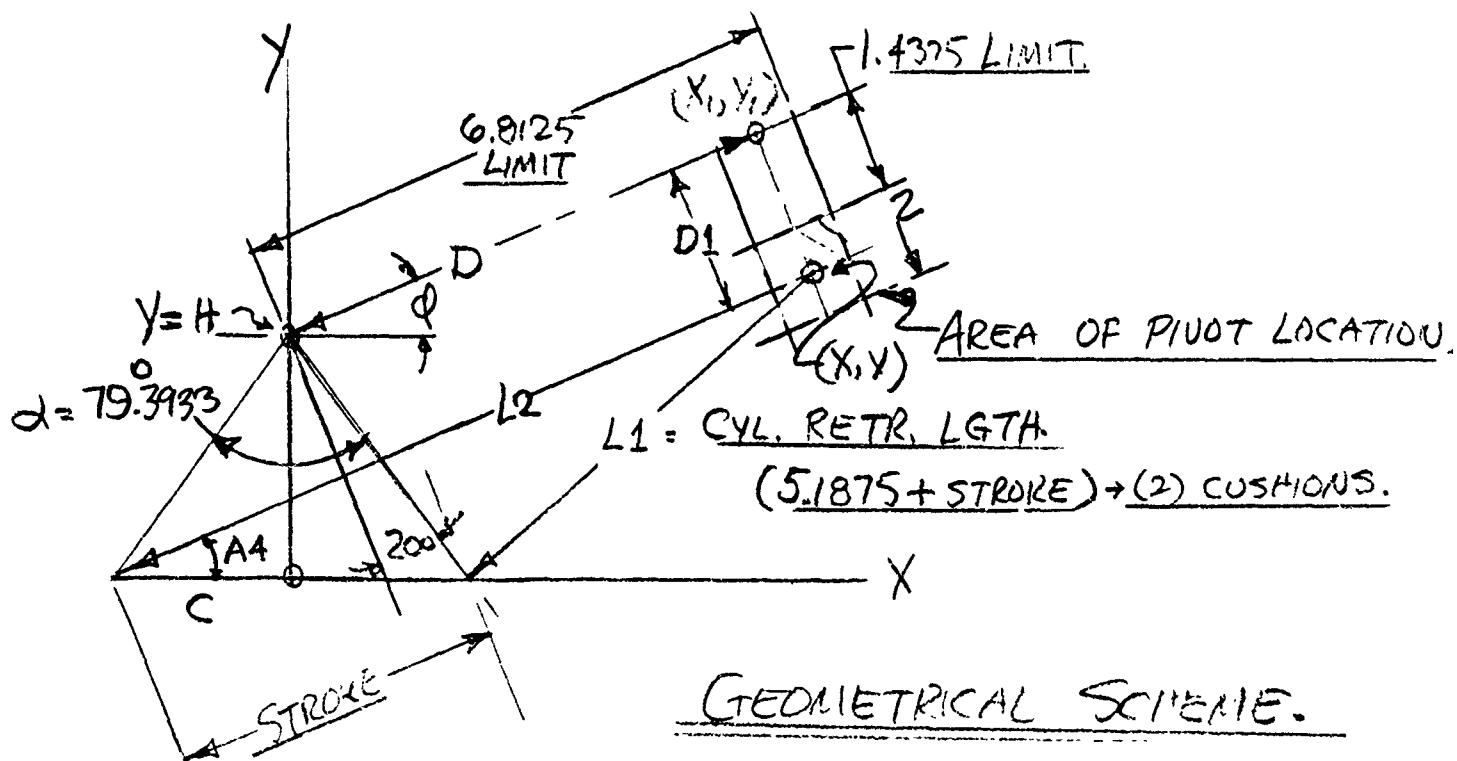
PAGE

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OF

POWDER TRAY SHIFT CYL.

PROBLEM: THE (SHIFT) CYLINDER SELECTED HAS STANDARD RODS, ETC., WITH STROKES IN $\frac{1}{4}$ IN. INCREMENTS. FROM LAY-OUTS & CALCULATIONS CERTAIN ANGLES & SPACE LIMITATIONS WERE DETERMINED. A COMPUTER PROGRAM WAS USED TO FIND THE ANCHOR POINT, STROKE, CRANK RADIUS, ETC., FOR A STD. CYL. WHICH WOULD OPERATE THE CRANK THRU THE DESIRED ANGLES.



PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME KBACE
DATE 12-3-79

REFERENCE A-L. RAMMER
PAGE 43 OF

SHIFT CYL. ATTACH. POINT

SEARCH PROGRAM.

KBAXIS1 08:33 12/03/79 MONDAY 106

```

1 DATA 20,79.3933 } LAYOUT DATA
2 DATA 5.1875
3 DATA 1,2.5,.25 } ESTIMATED LOOP LIMITS
4 DATA 1.25,2.25,.0625
10 READ E1,E2,S9
20 READ S1,S2,S3
30 READ R1,R2,R3
40 A9=E2/2-E1
50 A2=RAD(A9)
60 M1=TAN(A2)
70 M2=TAN(&PI/2+A2)
80 A3=RAD(E2)
90 PRINT 'STROKE','RADIUS','CHORD','L-COL.','L-EXT.','D','D1'
100 PRINT
110 FOR S=S1 TO S2 STEP S3 ← STROKE LOOP
120 L1=S+S9
130 L2=S+L1
140 FOR R=R1 TO R2 STEP R3 ← RADIUS LOOP
150 H=R*COS(A3/2)
160 C=2*SQR(R^2-H^2)
170 S0=(L1+L2+C)/2
180 IF S0<L2 THEN 330
190 R0=SQR((S0-L1)*(S0-L2)*(S0-C)/S0)
200 A4=2*ATN(R0/(S0-L1))
210 X=L2*COS(A4)-C/2
220 Y=L2*SIN(A4)
230 B2=Y-M2*X
240 X1=(B2-H)/(M1-M2)
250 Y1=M2*X1+B2
260 D1=SQR((X1-X)^2+(Y1-Y)^2)
270 IF D1<1.4375 THEN 320 } LIMITS (SEE SKETCH)
280 IF D1>2 THEN 320
290 D=SQR(X1^2+(Y1-H)^2)
300 IF D>6.8125 THEN 320 ← LIMIT
310 PRINT S,R,C,L1,L2,D,D1
320 IF CPU>9 THEN 360
330 NEXT R
340 NEXT S
350 GO TO 370
360 PRINT 'CPU>9.'
370 END

```

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME KBOWEE
DATE 12-3-79

REFERENCE A-L. POWER
PAGE 44 OF

SHIFT CYL. ATTACH POINT

KBAXIS1 08:35 12/03/79 MONDAY 106

STROKE	RADIUS	CHORD	L-COL.
1.25	1.3125	1.67665	6.4375
1.25	1.375	1.75649	6.4375
→ 1.5	1.5625	1.99601	6.6875 ←
1.5	1.625	2.07585	6.6875
1.75	1.8125	2.31537	6.9375
1.75	1.875	2.39521	6.9375
1.75	1.9375	2.47505	6.9375
2	2.25	2.87425	7.1875

L-EXT.	D	D1
7.6875	6.1921	1.67475
7.6875	6.03465	1.94505
→ 8.1875	6.49588	1.56213 ←
8.1875	6.35434	1.80455
8.6875	6.79439	1.45981
8.6875	6.66439	1.6806
8.6875	6.53676	1.87036
9.1875	6.73144	1.90012

PROGRAM PRINT-OUT OF POSSIBLE COMBINATIONS.

CHOSEN COMBO MARKED: 1.5" STROKE, 1.5625 CRANK

RADIUS, CYL. COLLAPSED LENGTH = 6.6875", CYL. EXT-

ENDED = 8.1875", D = 6.5", D1 = 1.56" ± TOLS.

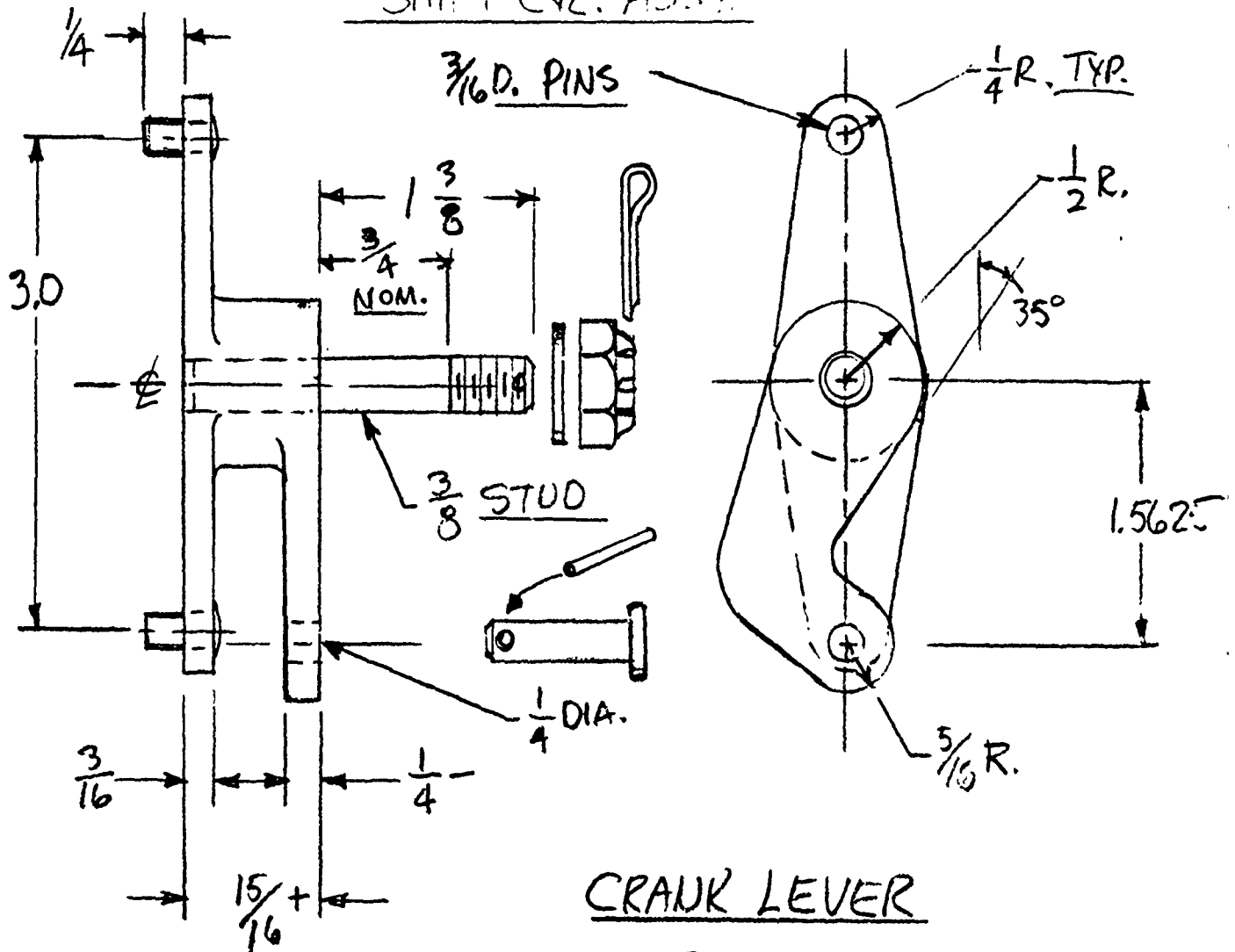
THIS ALLOWS FULL CYL. STROKE TO "MATCH" THE
FULL TRAY SLIDE ~ WITH CUSHIONED STOPS.

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME K. B. JEE
DATE 9-13-79

REFERENCE A-L, RAMMER
PAGE 45 OF

SHIFT CYL. ASSY.



CRANK LEVER

$$F_{ty} = 36 \text{ KSI STL.}$$

NOTE: PINS & STUD HAVE TO BE TIGHT-PEEVED,
EXPANDED, SOLDERED, WELDED, ETC

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME

K. Baez

REFERENCE

A-L RAMMER

DATE

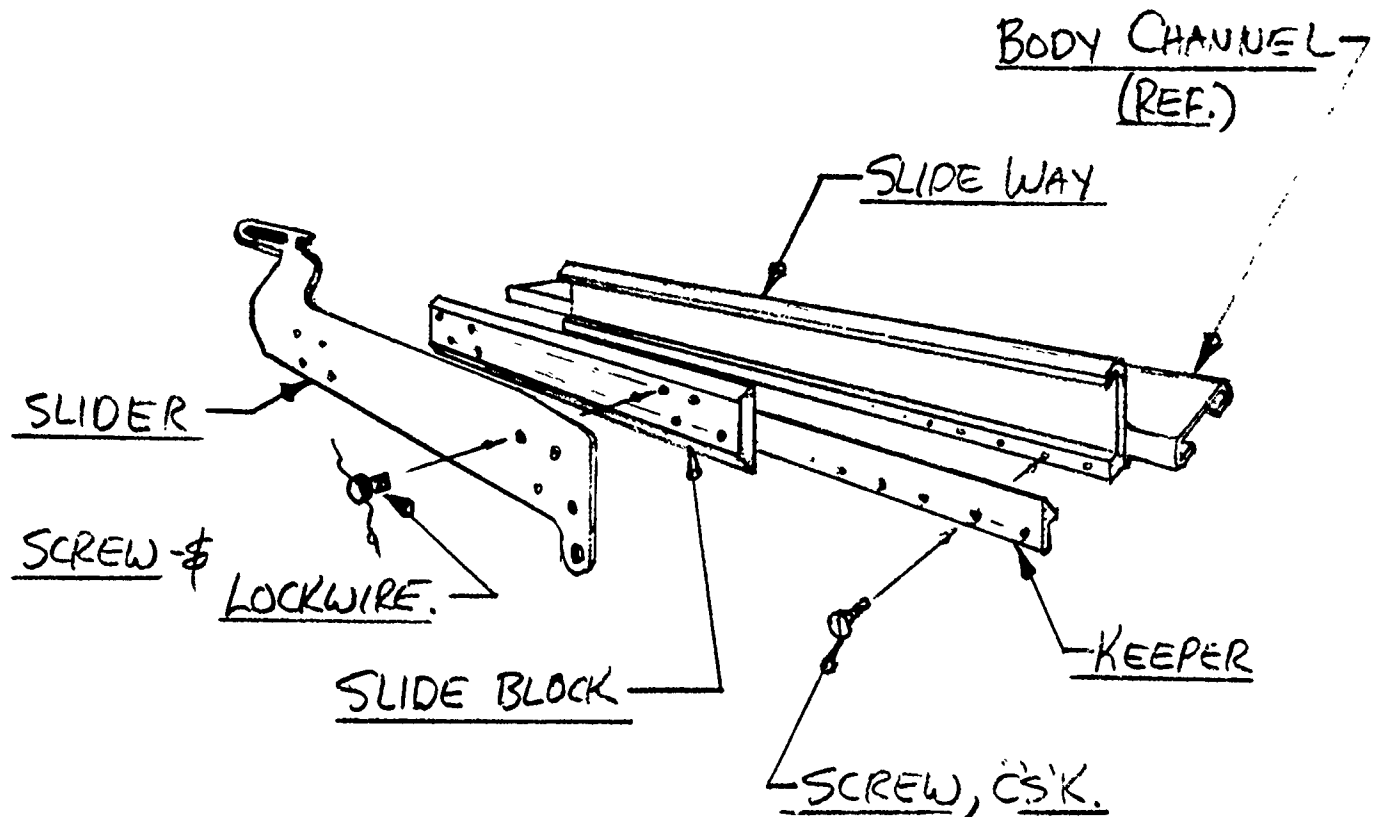
9-12-79

PAGE

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OF

SLIDE ASSEMBLY

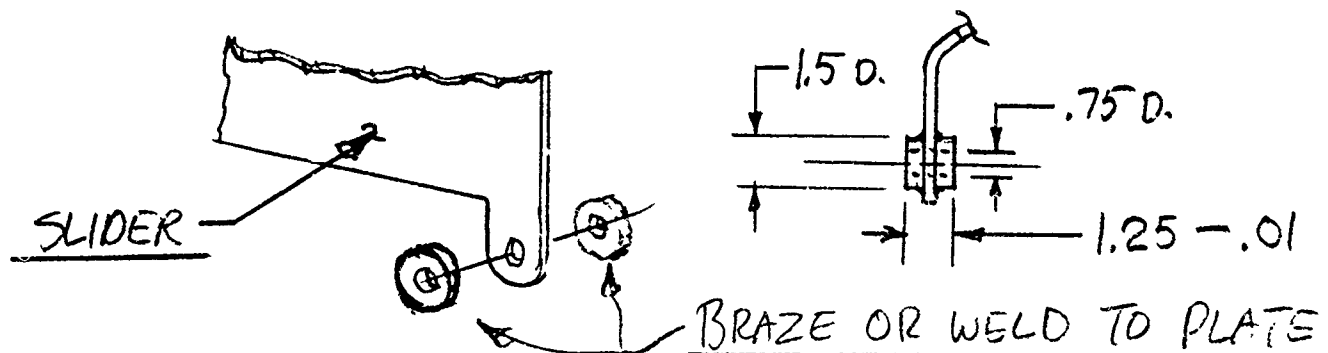


THE SLIDER IS $\frac{1}{8}$ " C. STEEL PLATE.

THE SLIDE BLOCK IS OILITE OR BRONZE.

THE SLIDE WAY IS $\frac{3}{16}$ C. STEEL SHEET.

THE KEEPER IS C. STEEL, $STL: F_{ty} = 36K.$ (OR BETTER.)



PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

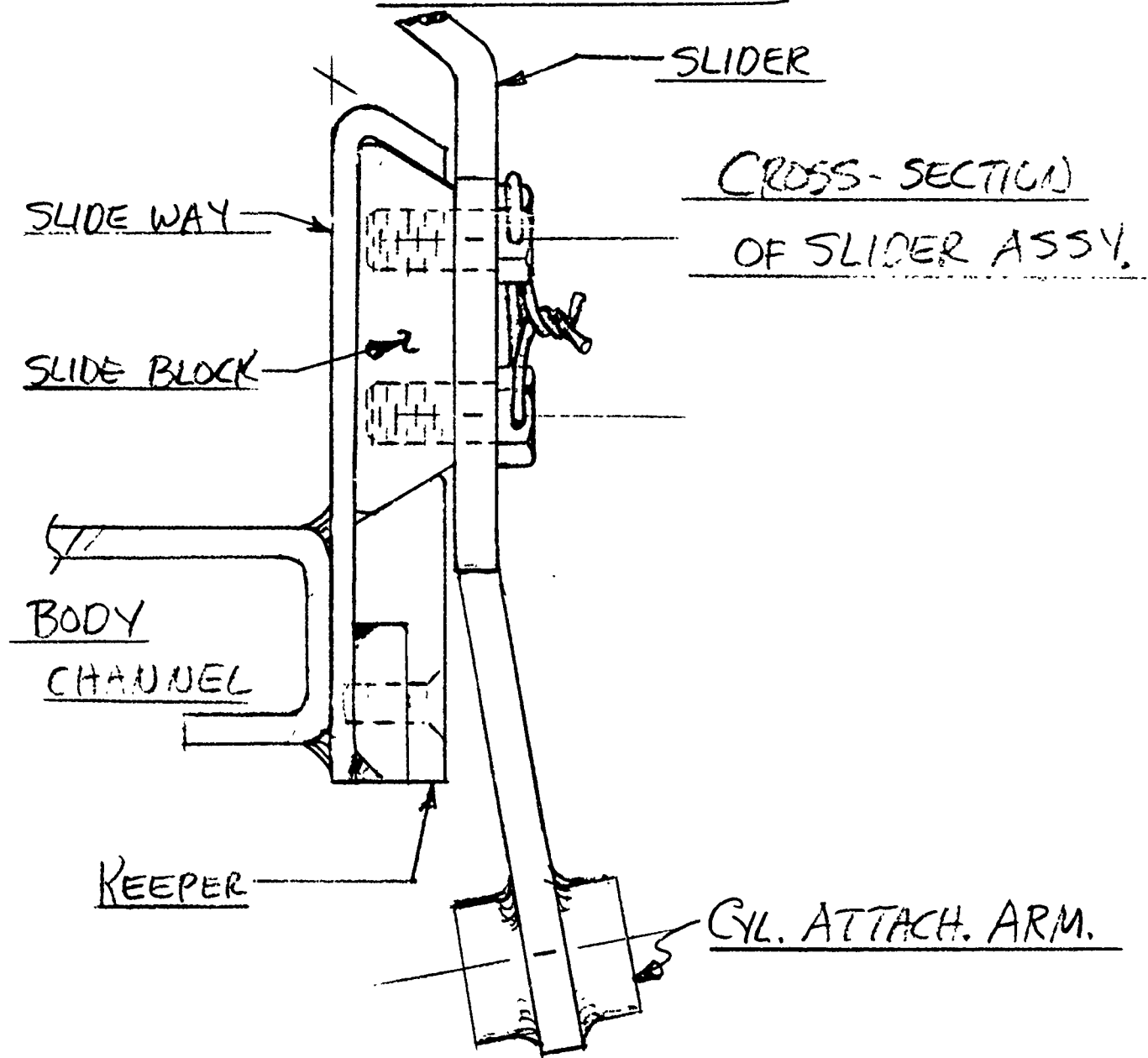
NAME K. B. ZEE

REFERENCE A-L RAMMER

DATE 9-12-79

PAGE 47 OF

SLIDE ASSEMBLY

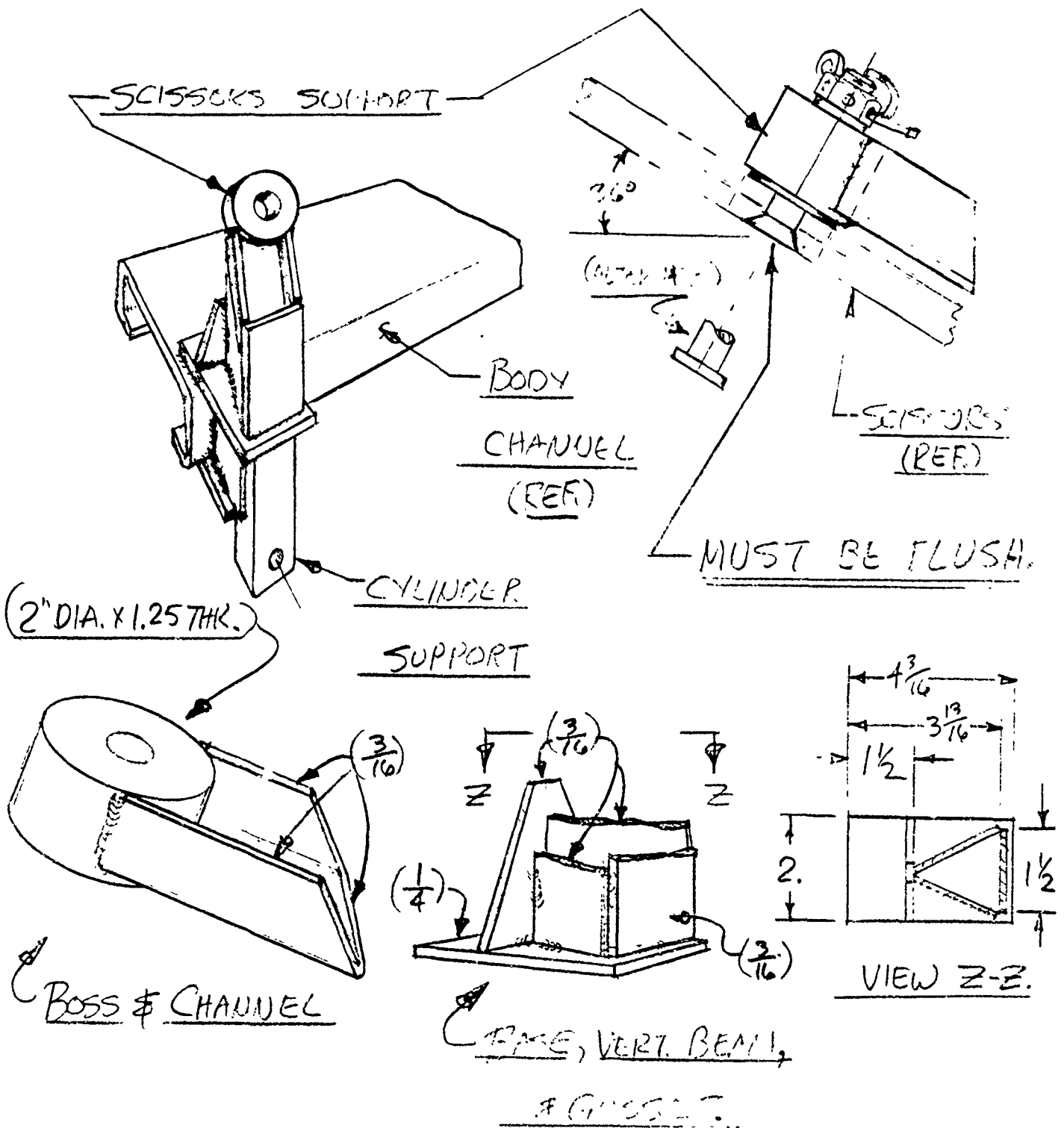


PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME K. B. FEE
DATE 9-12-79

REFERENCE A-1 RAMMER
PAGE 48 OF

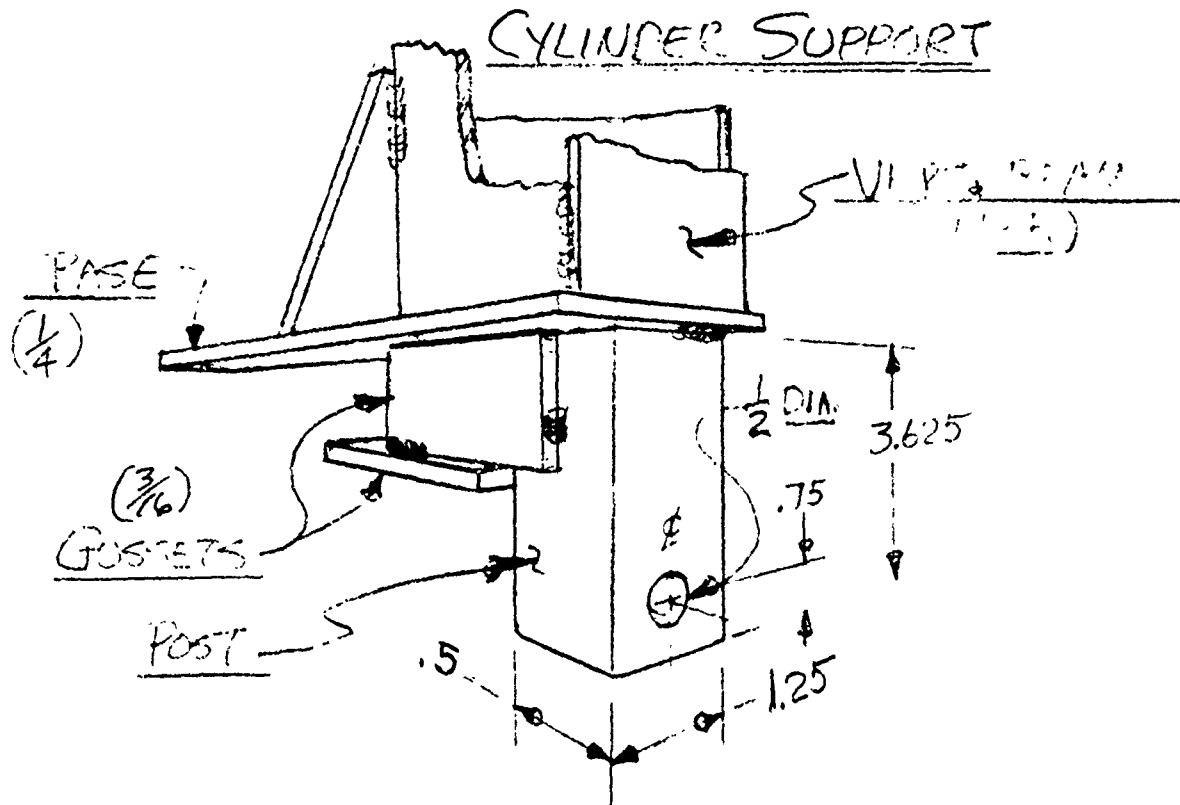
SCISSORS SUPPORT



PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME KB:EE
DATE 9-13-79

REFERENCE 1-1, ZAPPER
PAGE 49 OF



NOTE: THIS COULD BE A CATHODE EVENTUALLY.

ALL THE PARTS SHOWN IN THIS SECTION
AND ON THE LAYOUTS HAVE BEEN DESIGNED & DRAWN
TO BE HAND-MADE, CUT, WELDED, ETC., AS AN
EXPERIMENTAL MODEL. LATER DESIGNS COULD BE
MORE SOPHISTICATED.

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

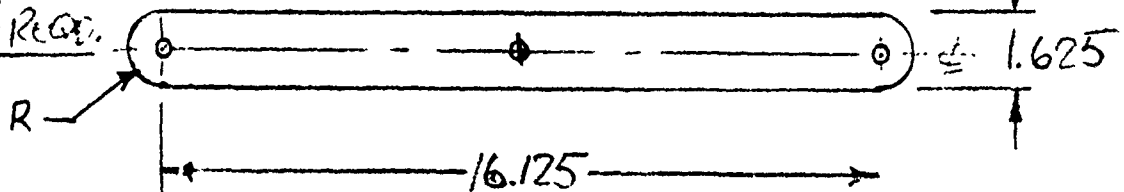
NAME K. BOUZZ
DATE 11-15-79

REFERENCE A-L. RAMMER
PAGE 50 OF

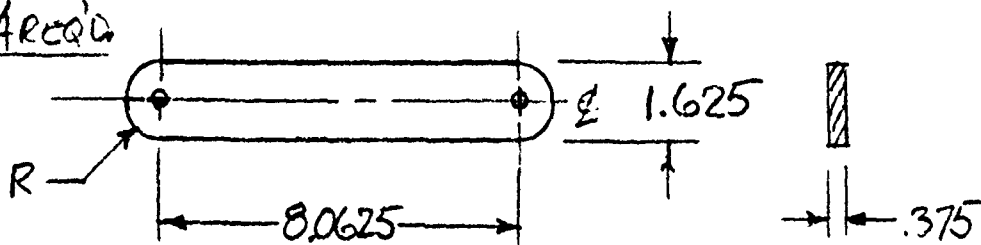
SCISSORS

LINKS:

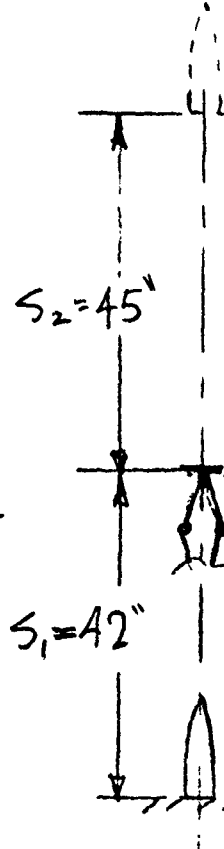
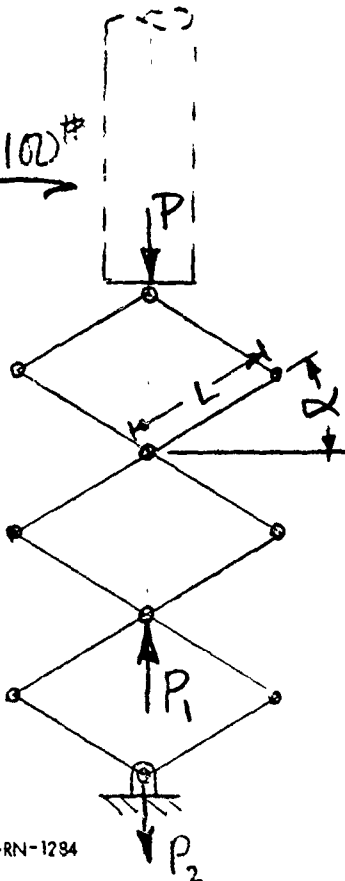
1) LONG: 4 REQ'D.



2) SHORT: 4 REQ'D.



WT = 100#



REQ'D 120 IN/SEC VELOCITY. = V_F

FREE THROW. (CALC. AS VERTICAL FOR MAX. NUMBER.)

VEL. REQ'D. = V_1 AT MAX. TRAVEL

RAMMER TRAVEL

$$V_1 = \sqrt{V_F^2 + 2gS_2}$$

FORMULA

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME KBUEZ
DATE 11-19-79

REFERENCE A-L. RAMMER
PAGE 51 OF

SCISSORS
CALCS.

$$V_1 = \sqrt{120^2 + 2(386)45} = 221.67544 \text{ IN/SEC}$$

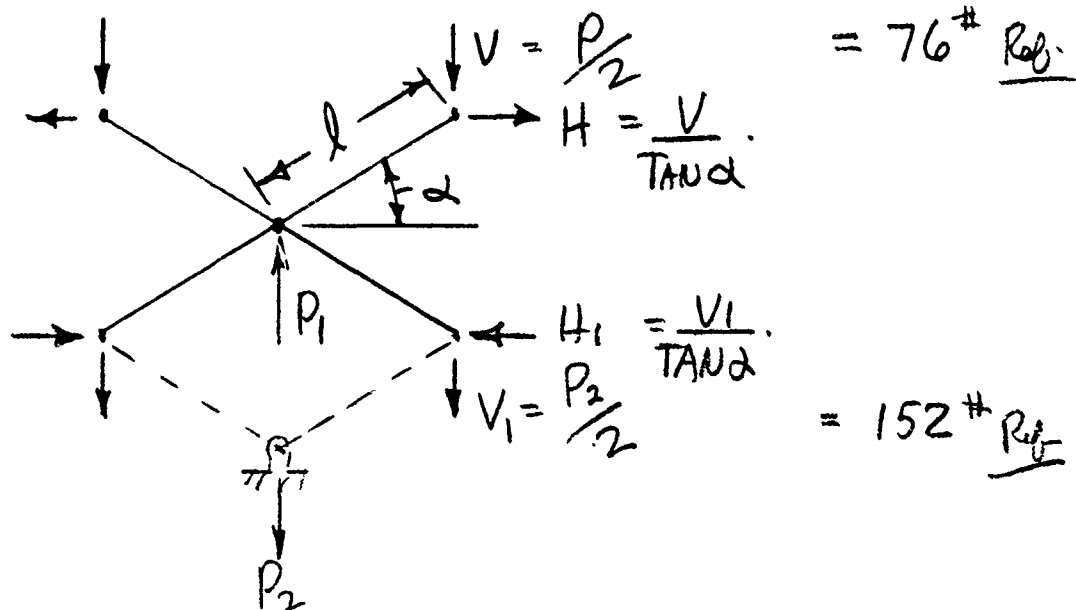
$$P = \frac{WV_1^2}{29S_1} = \frac{100(221.67544)^2}{2(386)42} = 151.55441 \text{ #}$$

USE 152*

LINKAGE HAS MECH. ADVANTAGE OF -3.

$$P(42) = P_1(14) ; P_1 = \frac{152(42)}{14} = 456 \text{ #}$$

$$P_1 - P = P_2 = 304 \text{ #}$$



NAME

KBOZZE

REFERENCE

A-L. RIMMER

DATE

11-19-79

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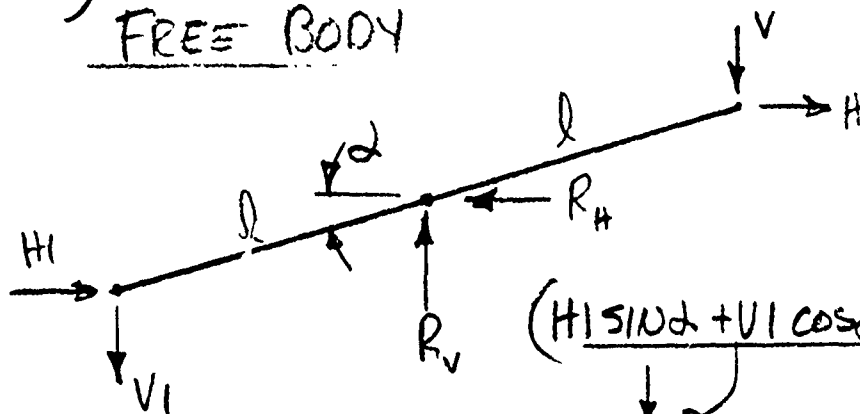
OF

SCISSORS

CALCS.

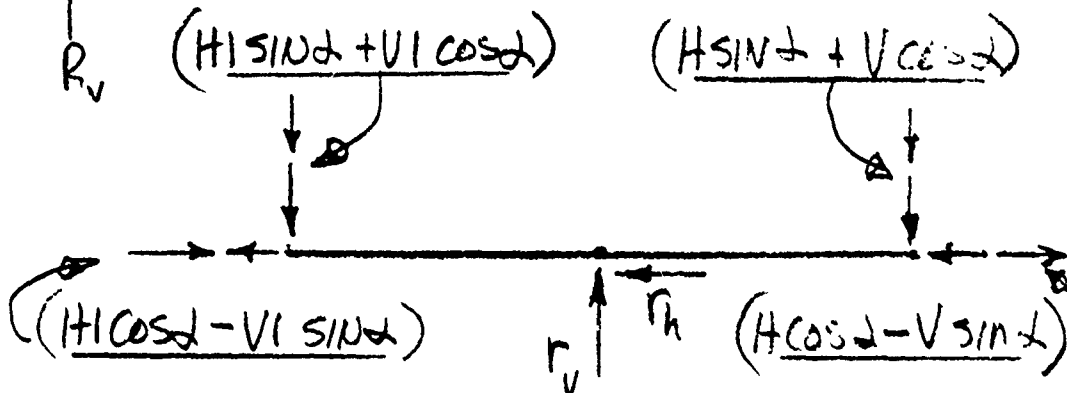
1)

FREE BODY



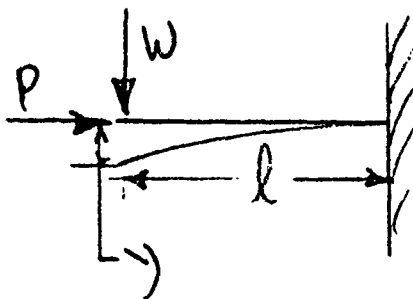
2)

ROTATE



3)

CALC. LINK AS TWO CANTILEVER BEAMS



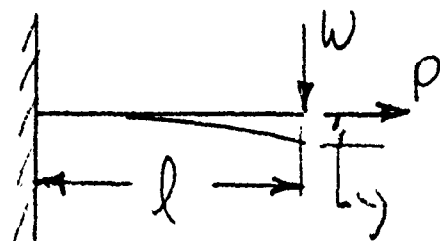
USE ROARK

FORMULAS:

$$J = \sqrt{\frac{EI}{P}}; U = \frac{l}{J}$$

$$M = -WJ \tanh(U)$$

$$y = -\frac{W}{P} (J \tanh(U) - l)$$



$$M = -WJ \tanh(U)$$

$$y = -\frac{W}{P} (l - J \tanh(U))$$

MAX. MOM. & STRESS WOULD BE AT MIN. (alpha).

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME

KC

DATE

11-20-79

REFERENCE

A-L P

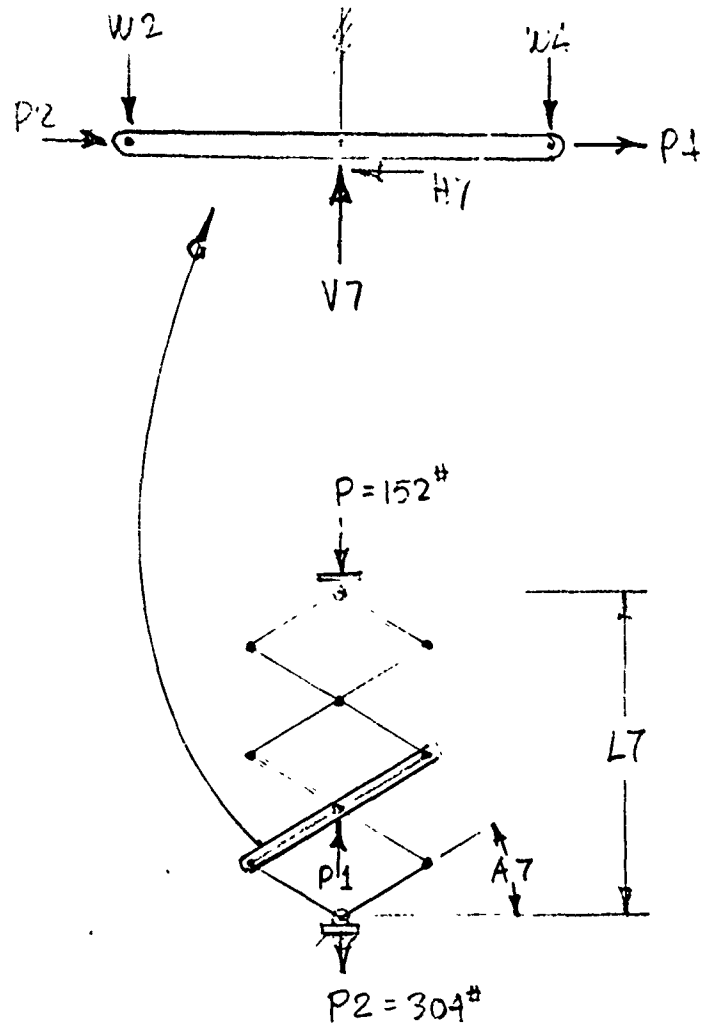
PAGE

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OF

SC
CH

KBSCISR2	W2 #	P2 #	W4 #	P4 #	V7 #	H7 #	A7 °	L7 in.
297.022	548.752	21.337	148.511	-324.376	445.533	973.128	12.3000	10.3053
290.250	420.793	-16.506	145.125	-210.397	435.375	631.190	17.2983	14.3842
281.270	285.292	-48.347	140.635	-142.646	421.906	427.937	22.2967	18.3536
270.152	192.058	-75.118	135.076	-96.029	405.228	288.087	27.2950	22.1834
256.978	122.096	-97.417	128.489	-61.048	385.468	183.143	32.2934	25.8445
241.851	66.692	-115.629	120.925	-33.346	362.776	100.039	37.2917	29.3091
224.884	21.337	-130.006	112.442	-10.668	337.325	32.005	42.2900	32.5507
206.206	-16.506	-140.712	103.103	8.253	309.309	-24.760	47.2884	35.5448
185.960	-48.347		92.980	24.173	278.940	-72.520	52.2867	38.2686
164.300	-75.118		82.150	37.559	246.450	-112.678	57.2850	40.7012
141.390	-97.417		70.695	48.709	212.085	-146.126	62.2834	42.8244
117.405	-115.629		58.703	57.615	176.108	-173.444	67.2817	44.6218
92.527	-130.006		46.264	65.003	138.791	-195.009	72.2800	46.0799
66.945	-140.712		33.473	70.356	100.418	-211.068	77.2784	47.1875



LOADS ANALYSIS.

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME K. B. BOE
DATE 11-19-79

REFERENCE A-L. RAMMER
PAGE 54 OF

SCISSORS

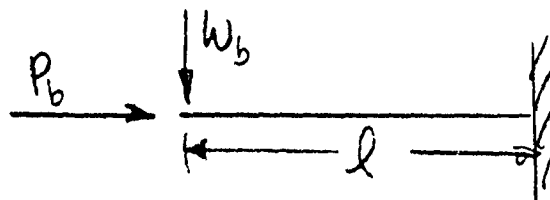
CALCS.

MIN. $\alpha = 12.3^\circ$ PER LAY-OUT.

$V_1 = 304/2 = 152^\#$; $H_1 = 697.135^\#$

$W_b = H_1 \sin \alpha + V_1 \cos \alpha = 297.022^\#$

$P_b = H_1 \cos \alpha - V_1 \sin \alpha = 648.752^\#$



$I = \frac{.375(1.625^3)}{12}$

$E = 29,000,000$

$M = -297.022(77.422056)(.1045150785)$

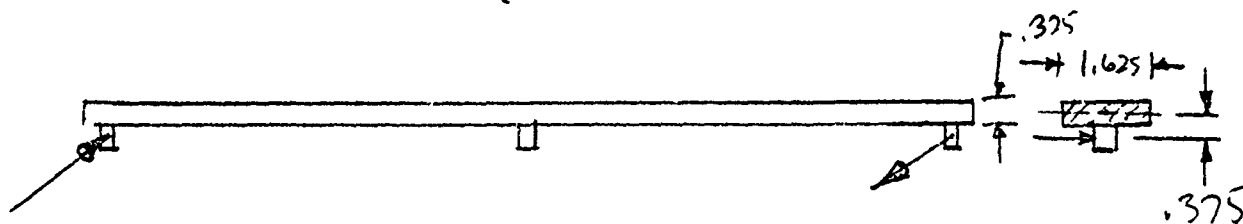
$M = -2403.4344^\#$

$J = \sqrt{\frac{EI}{P}} = 77.422$

STRESS $= \frac{M}{I} = 14,562.19563 \text{ psi}$

$U = \frac{1}{J} = .104137$

$\delta = -\frac{297.022}{648.752} (77.422056(.1045150785) - 8.0625) = -.0139$



TORSIONAL MOMENT $= W(.375) = 111.38325^\#$

$\tau = \frac{111.38325}{.283(1.625)(.375^2)} = 1722.3347 \text{ psi}$ TORS. SHEAR ϕ LONG SIDE

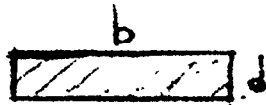
PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME H. Boe
DATE 11-19-79

REFERENCE A-L. RIVIER
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SCISSORS

CALCS.



$$R = \beta b d^3 = .282 (1.625) (.975)^3$$

$$R = .02416553 ; \theta = \frac{TL}{GR} = \frac{111.38325 (16.125)}{1150000 (.02416553)}$$

$$\theta = .0064628703 \text{ RAD.} = .3702952 \text{ DEGREES}$$

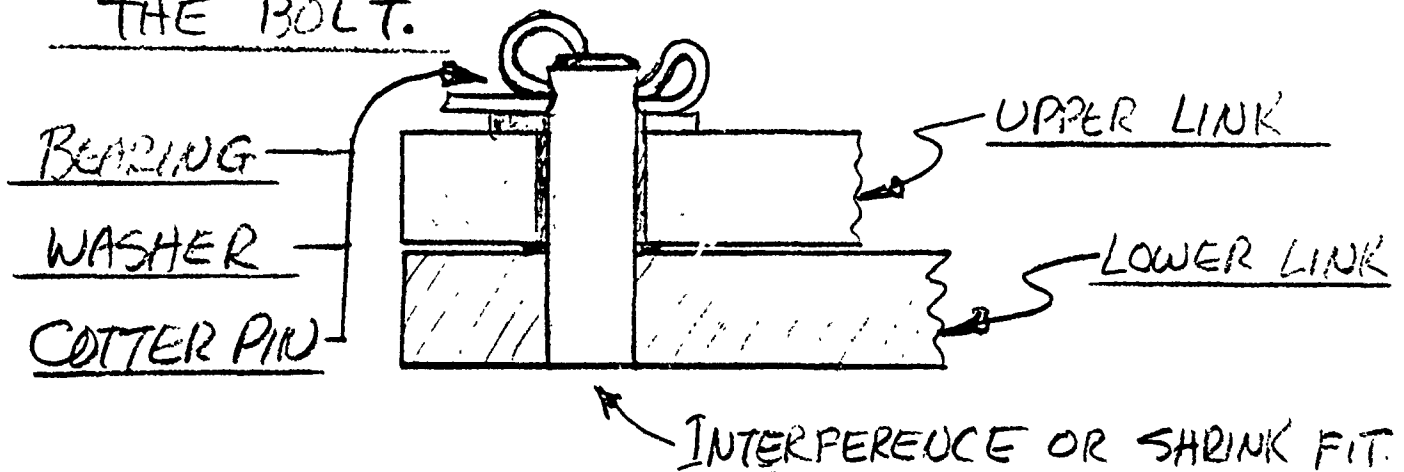
WORST CASE! (0° 22' 13".063)

ACTUALLY, THE ABOVE CASE COULD

ONLY HAPPEN IF THE HOLE IN THE LOADING

PART WAS NOT TIGHT ENOUGH TO CONSTRAIN

THE BOLT.



LINK END BOLTS, ETC.

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME

KBOUKE

REFERENCE

A-L. RAMMER

DATE

11-19-79

PAGE

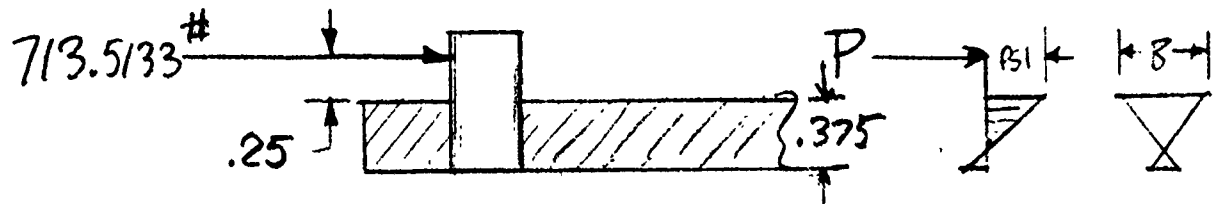
56

OF

SCISSORS

CALCS.

STUD OR BOLT BEARING STRESS IN HOLE.



KBEARSTR 14:38 11/19/79 MONDAY 106

THIS PROG. CALCS. BEARING STRESS IN
HOLES DUE TO BOLT BEARING & MOMENT.

ENTER BOLT (SHAFT) & HOLE DIAMETERS.

? .369, .375

ENTER HOLE DEPTH, BOLT LOAD & MOMENT ARM.

? .375, 713.5133, .25

ENTER POISSONS RATIO & "E" FOR BOLT MATERIAL.

? .29, 29500000

ENTER POISSONS RATIO & "E" FOR HOLE MATERIAL.

? .29, 29500000

ENTER HOLE DEPTH INCREMENT (INCHES) FOR LOOP LIMIT.

? .03125

NO.	STRESS-ROARK	psi.	B-ROARK	10.	B-ALT.	10.
1	65041.6		.186755		.186755	
2	60550.4		.173859		.173859	
3	55698.3		.159927		.159927	
4	50381.		.14466		.14466	
5	44431.9		.127578		.127578	
6	37551.8		.107823		.107823	
7	29087.5		8.35193E-02		8.35193E-02	
8	16793.7		4.82201E-02		4.82201E-02	
9	16793.6		4.82198E-02		4.82198E-02	
10	29087.5		8.35192E-02		8.35192E-02	
11	37551.8		.107823		.107823	
12	44431.9		.127578		.127578	
13	50381.		.14466		.14466	

THE FOLLOWING PRINT-OUTS ARE FOR (2) FAST METHODS FOR MAX. BRG. STRESS CALCS.

ROARK "B" P/A= 10188.2 PSI M/S= 54337.2 PSI TOTAL= 64525.5

B-ALT. P/A= 10188.2 M/S= 54337.2 TOTAL= 64525.5

SECTION MOD.= 3.28280E-03 FOR ROARK "B" & 3.28280E-03 FOR B-ALT. PSI.

NOTE: IF ROARK "B"< BOLT DIAM., PRINT-OUTS WILL BE EQUAL.

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

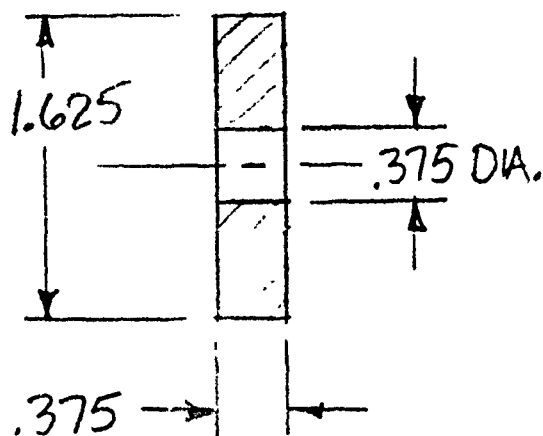
NAME K. BOVEE
DATE 11-19-79

REFERENCE A-L. RAMMER
PAGE 57 OF

SCISSORS.

CALCS.

RECALC. OF LINK STRESS ON SECTION
AT BOLT HOLE.



$$I = .132446 \text{ IN}^4$$

$$A = .46875 \text{ IN}^2$$

$$\text{STRESS} = 14,743.6 \text{ PSI.}$$

(COMPUTER CALCS.)

COMMENT:

THE LOADS & STRESSES, ETC.,
ON THE SCISSORS LINK SO FAR HAVE BEEN
ACTUAL - NO F.S. OR IMPACT ALLOWANCES,
NO PARTICULAR STEEL HAS BEEN SUGGESTED
USING 1020-25 STEEL BAR, THE BASIC

FACTORS OF SAFETY ARE: TENSILE $-\frac{36000}{14743.6} = 2.44;$

SHEAR $\frac{36000(.72)}{2453.33} = 10.565$; BEARING $-\frac{90000}{64525.5} = 1.395$

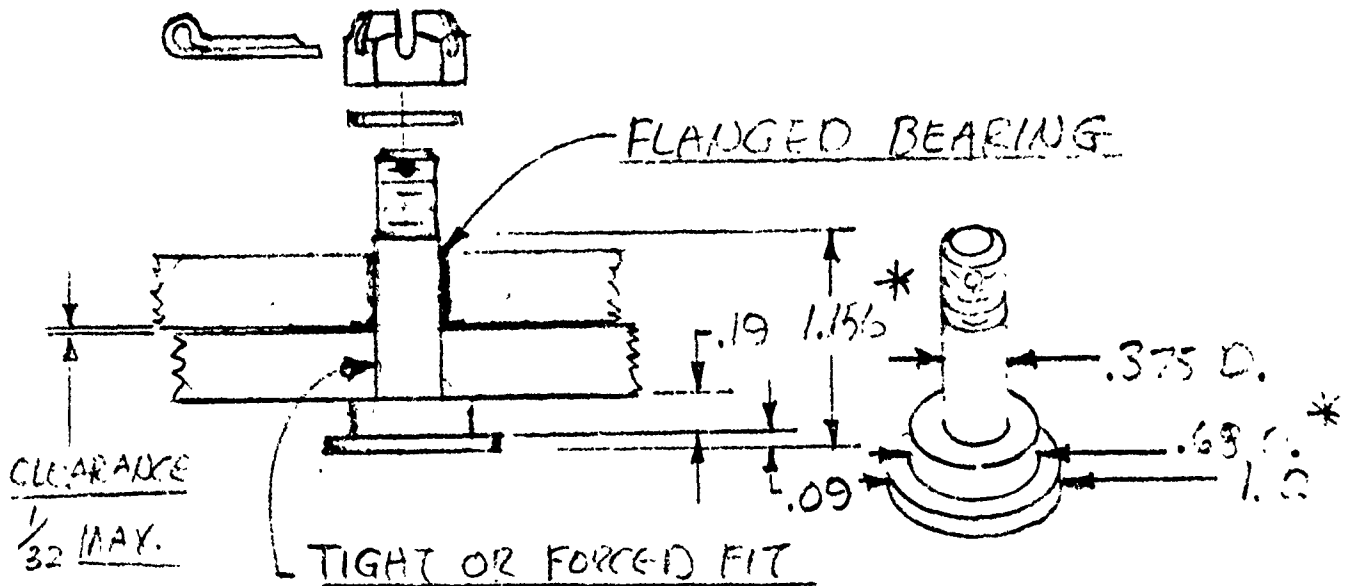
PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME K. BOE
DATE 11-20-77

REFERENCE A-L. RAMMER
PAGE 58 OF

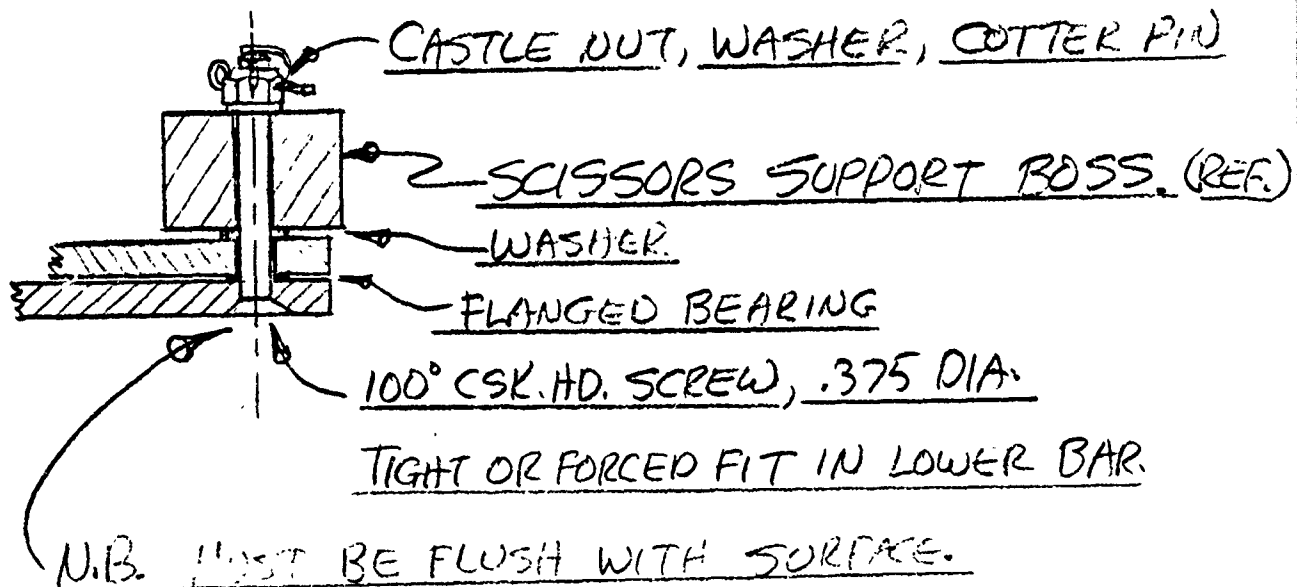
SCISSORS

AXIS BOLT



TWO (2) BOLTS REQ'D.

REAR SCISSORS POST



PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME

K. BOSEE

REFERENCE

A-L RAUNER

DATE

11-20-79

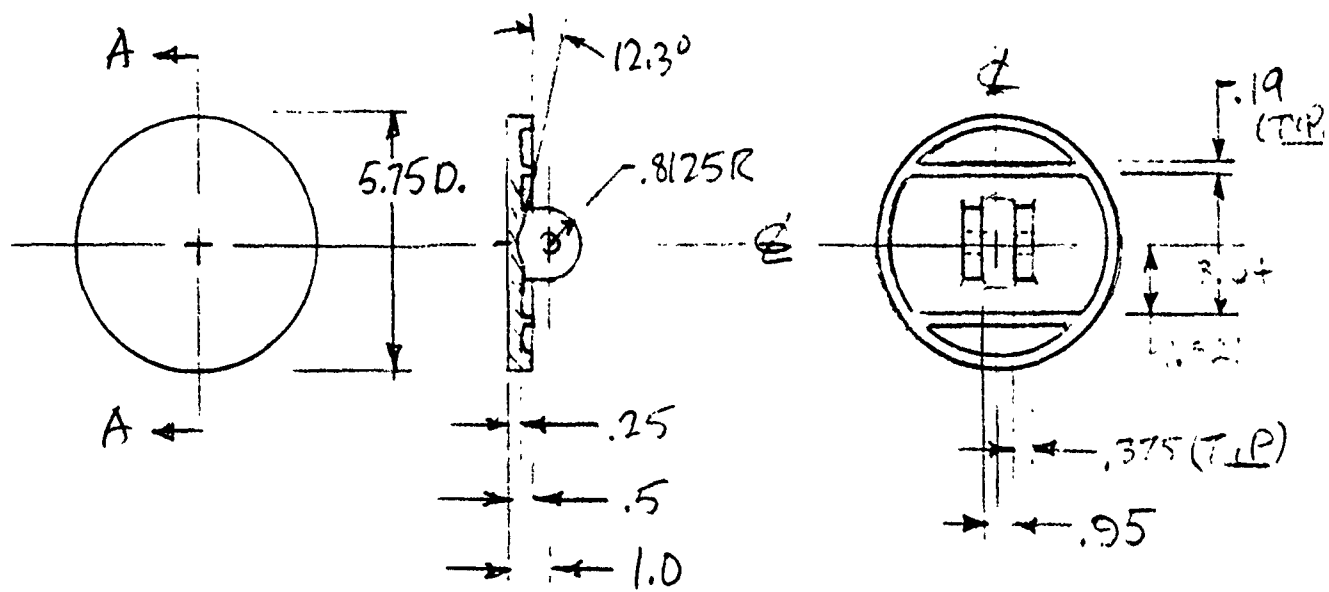
PAGE

59

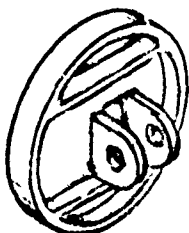
OF

SCISSORS.

PUSH PLATE ASSY.



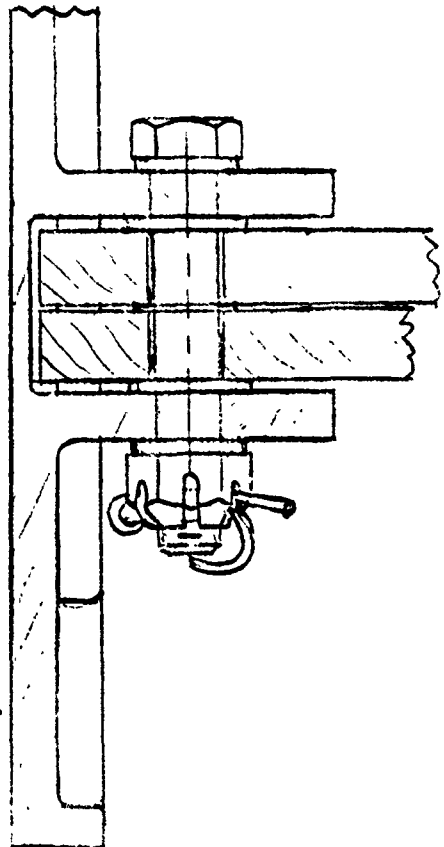
SECT. A-A



ISOMETRIC VIEW

90° ROTATED SECTION

SHOWING BOLT, ETC.



PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

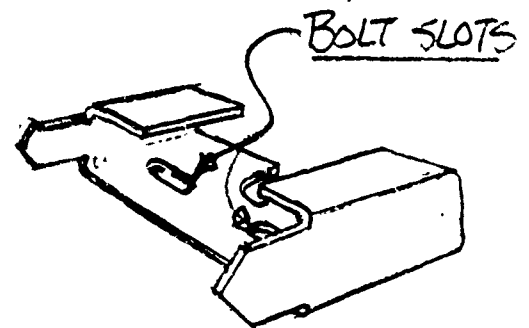
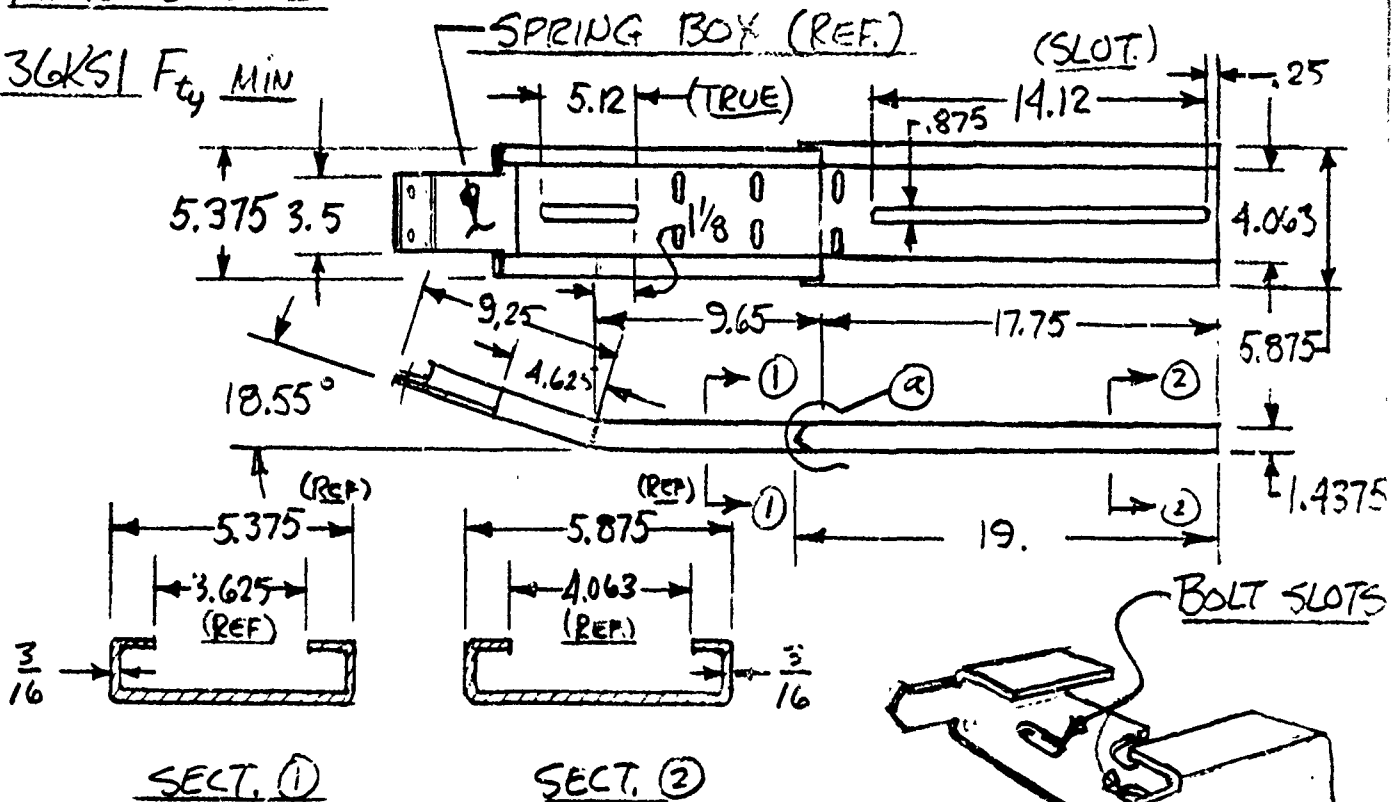
NAME K BOVEE
DATE 9-6-79

REFERENCE A-L RAMMER
PAGE 60 OF

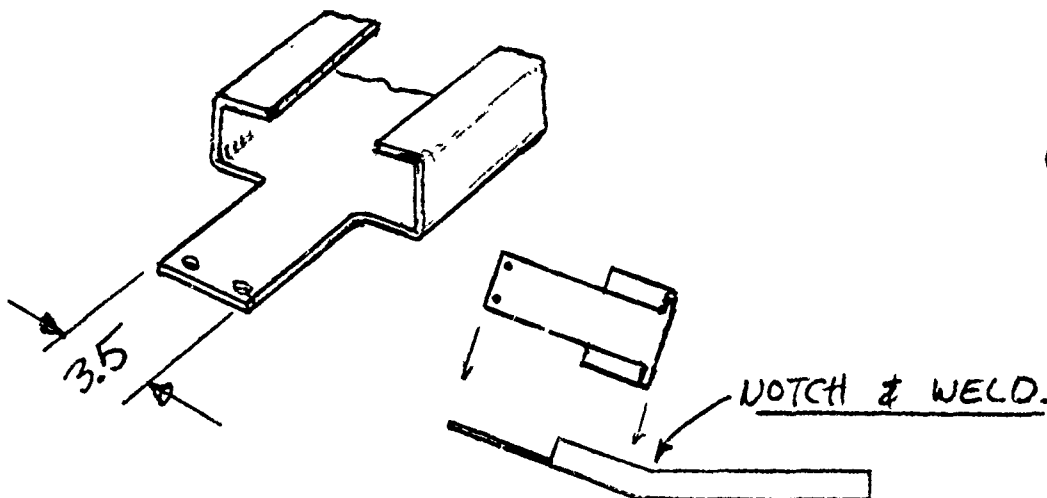
SUPPORT CHANNEL
WELD ASSEMBLY

MAT. STEEL

36KSI F_{ty} MIN



DETAIL ①
(Channel Splice)



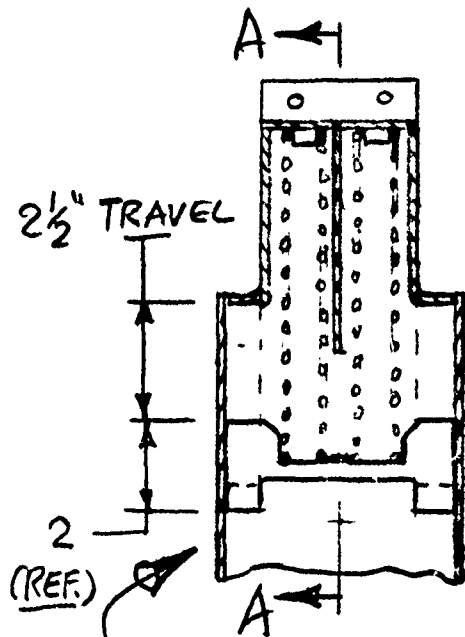
CHANNEL END DETAIL

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

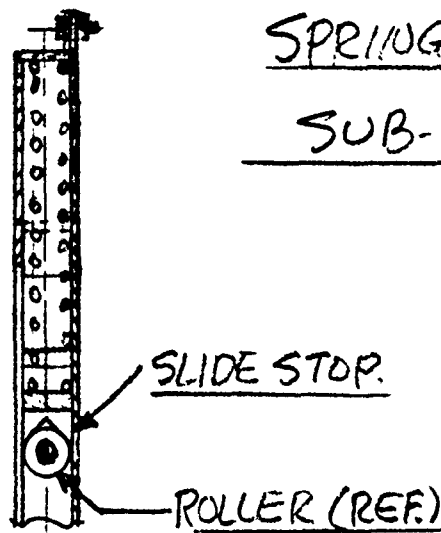
NAME KBOZZE
DATE 9-6-79

REFERENCE A-L RAMMER
PAGE 61 OF

SUPPORT CHANNEL
ASSEMBLY.



VIEW WITH
TOP SHEET REMOVED:

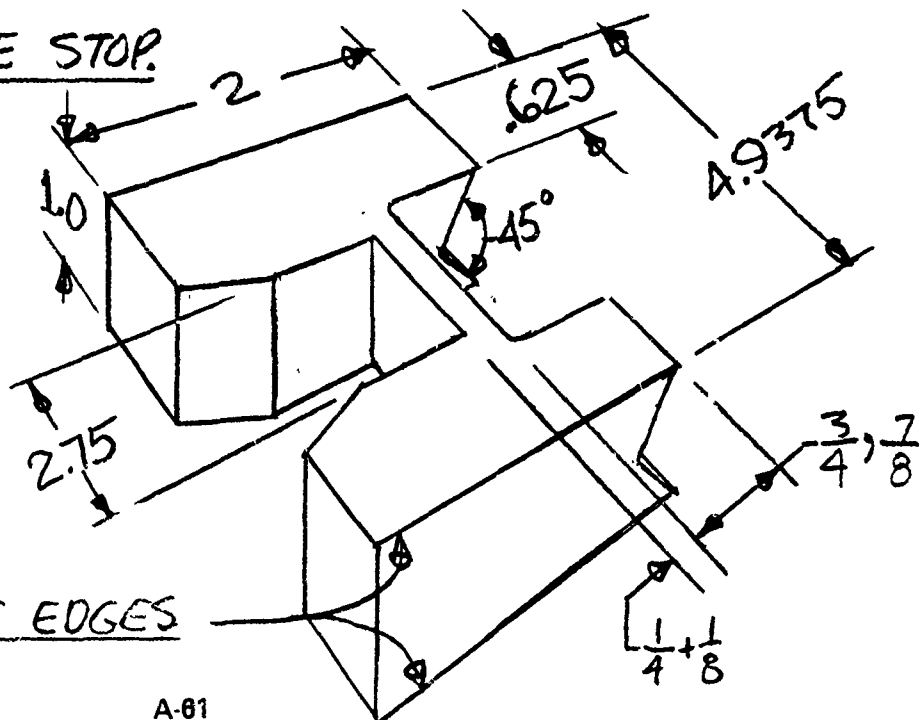


SECT. A-A.

SPRING BOX
SUB-ASSEMBLY.

SHOWS (2) SPRINGS,
SPRING BOX, & SLIDE STOP.

SLIDE STOP



ROUND OFF EDGES
(TYP.)

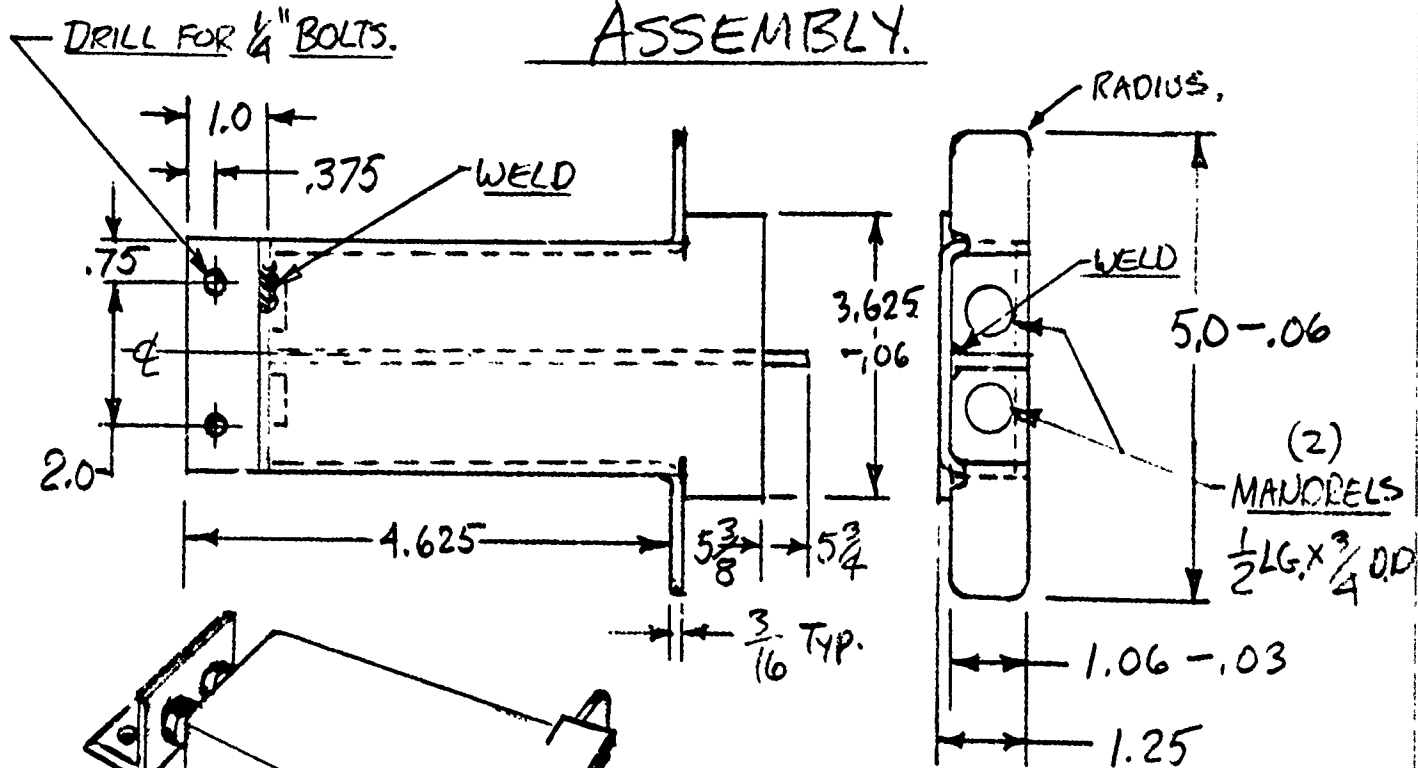
PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME KBOVEE
DATE 9-6-79

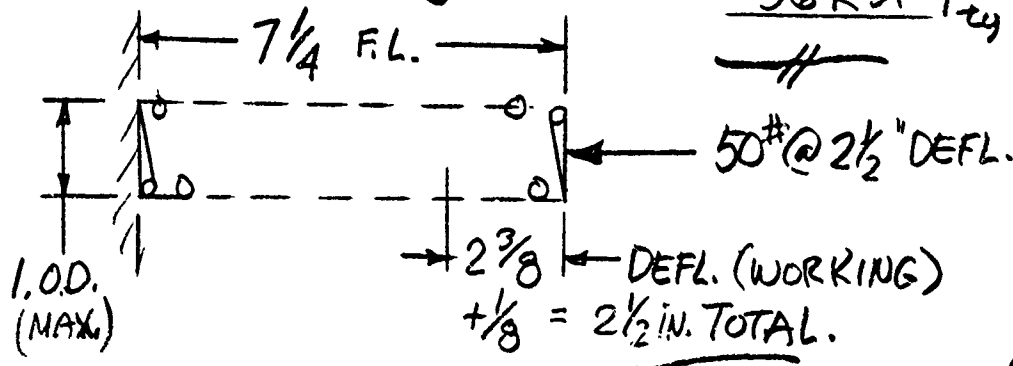
REFERENCE A-L RANNER
PAGE 62 OF

SUPPORT CHANNEL

ASSEMBLY.



SPRINGS (2)



$$N = \frac{2.5(11500000)(.106^4)}{8(50).9^3} = 12.45$$

$$S = \frac{8(50).9(1.17)}{\pi(.106)^3} = 112,569.54 \text{ PSI}$$

PCF-RN-1284

A-62

SPRING BOX DETAILS

MAT. STEEL SHEET. (3/16)

36 KSI F_{ty} MIN.

CALCS.

$$D = 1-d$$

$$\text{ASSUME } \frac{D}{d} = 8$$

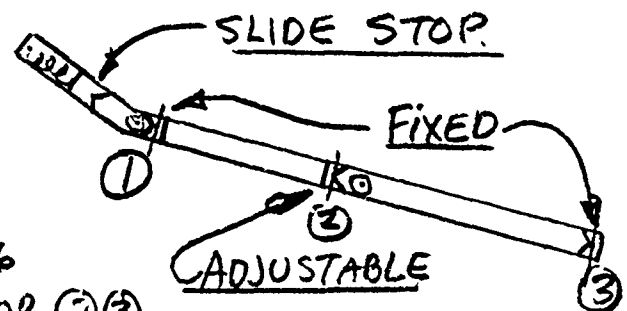
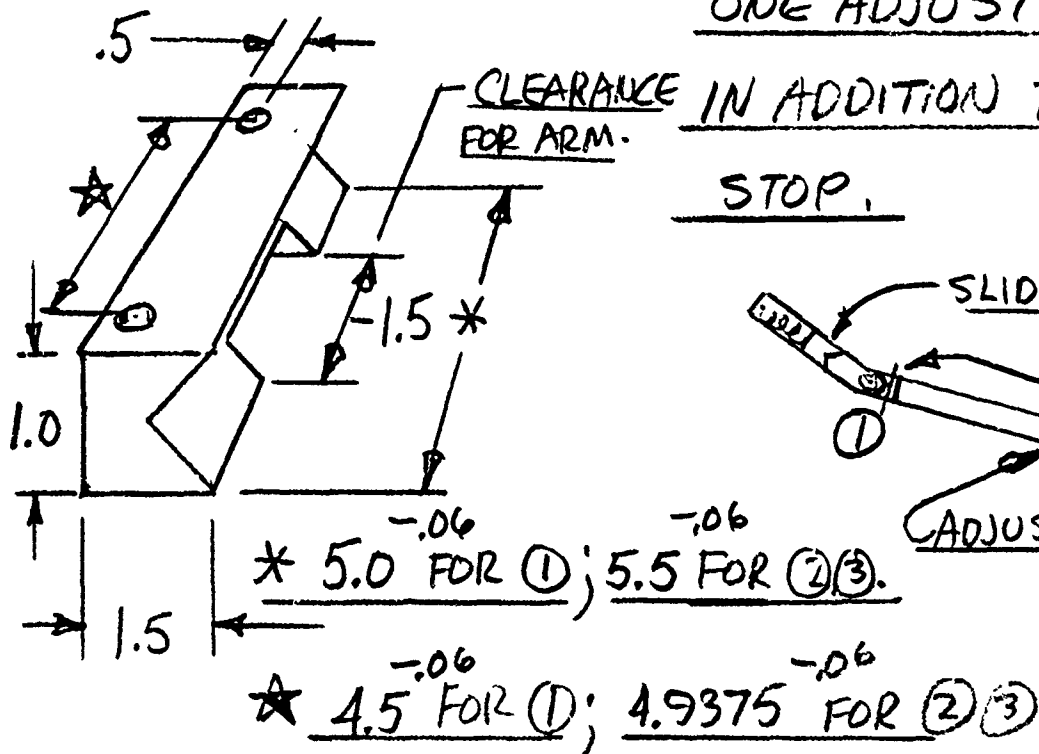
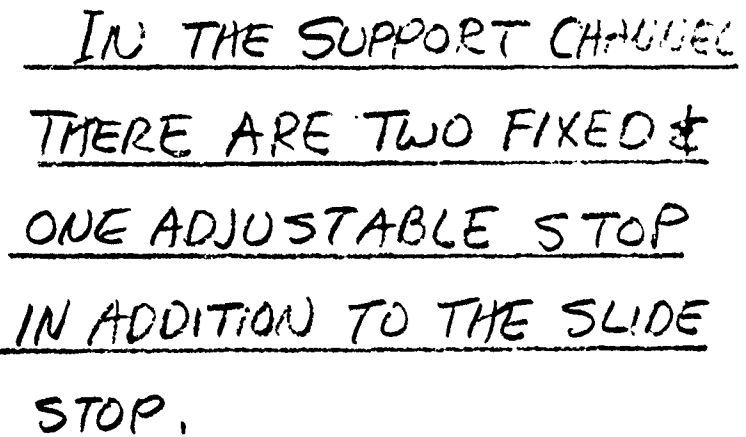
$$d = \frac{1}{9} = .111$$

USE .106 (35 GA.)
MUSIC WIRE

$$\frac{D}{d} = \frac{.9}{.106} = 8.49$$

$$K = \frac{1.17}{4}$$

TRAVEL STOPS:



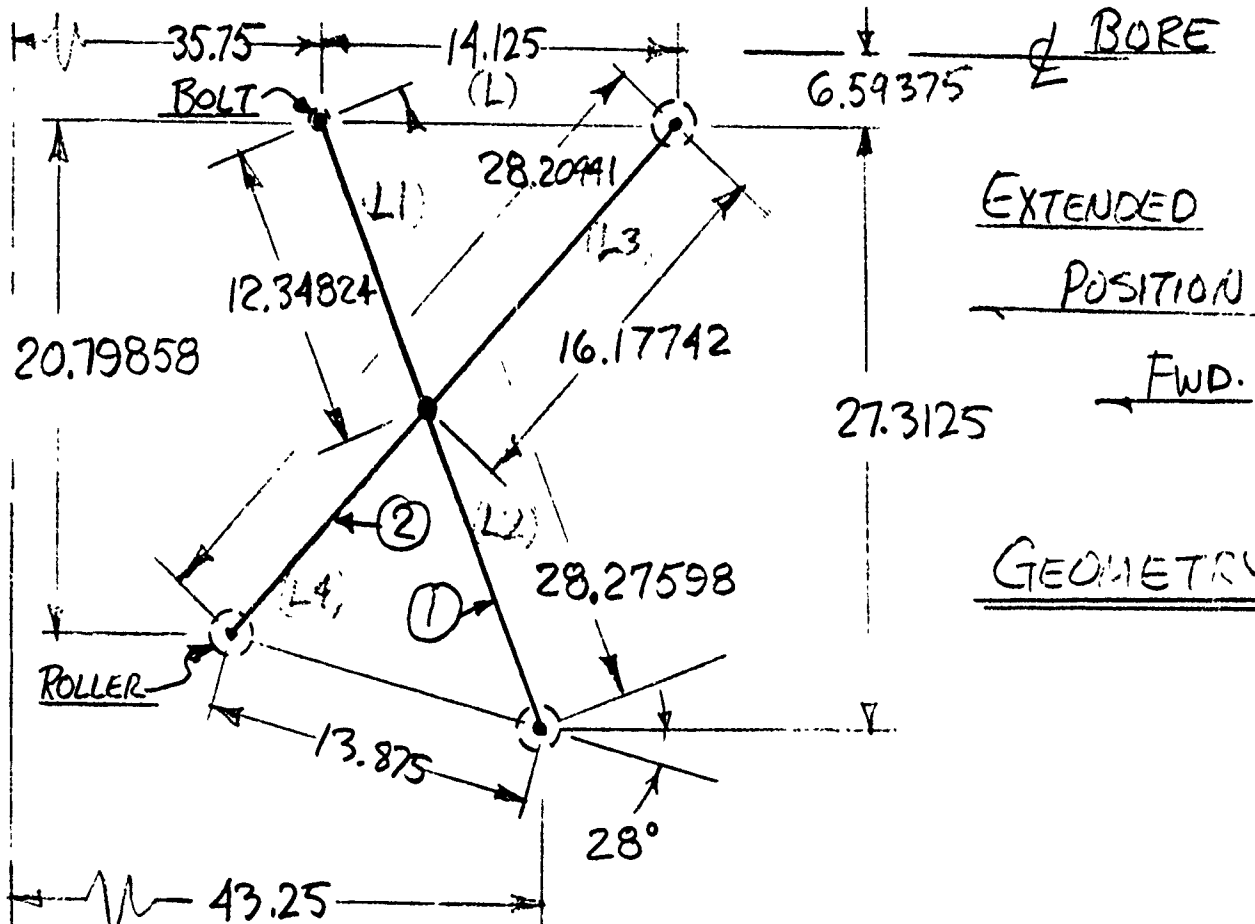
PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME K. BOREE
DATE 9-18-79

REFERENCE A-L. RANMER
PAGE 64 OF

TRUNNION

ERECTOR CHANNELS



NOTES: 1) DIMENSIONS ARE COORDINATED BUT NOT FINAL. INDIVIDUAL LAYOUTS WILL DETERMINE EXACT DIMENSIONS & TOLERANCES.

2) WELDS HAVE BEEN PICTORIALY INDICATED ON SOME SKETCHES BUT NOT CALLED OUT. ADJACENT METAL THICKNESSES (TO BE WELDED) HAVE BEEN SELECTED WITH WELDING IN MIND.

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME

H. Bovee

DATE

9-19-79

REFERENCE

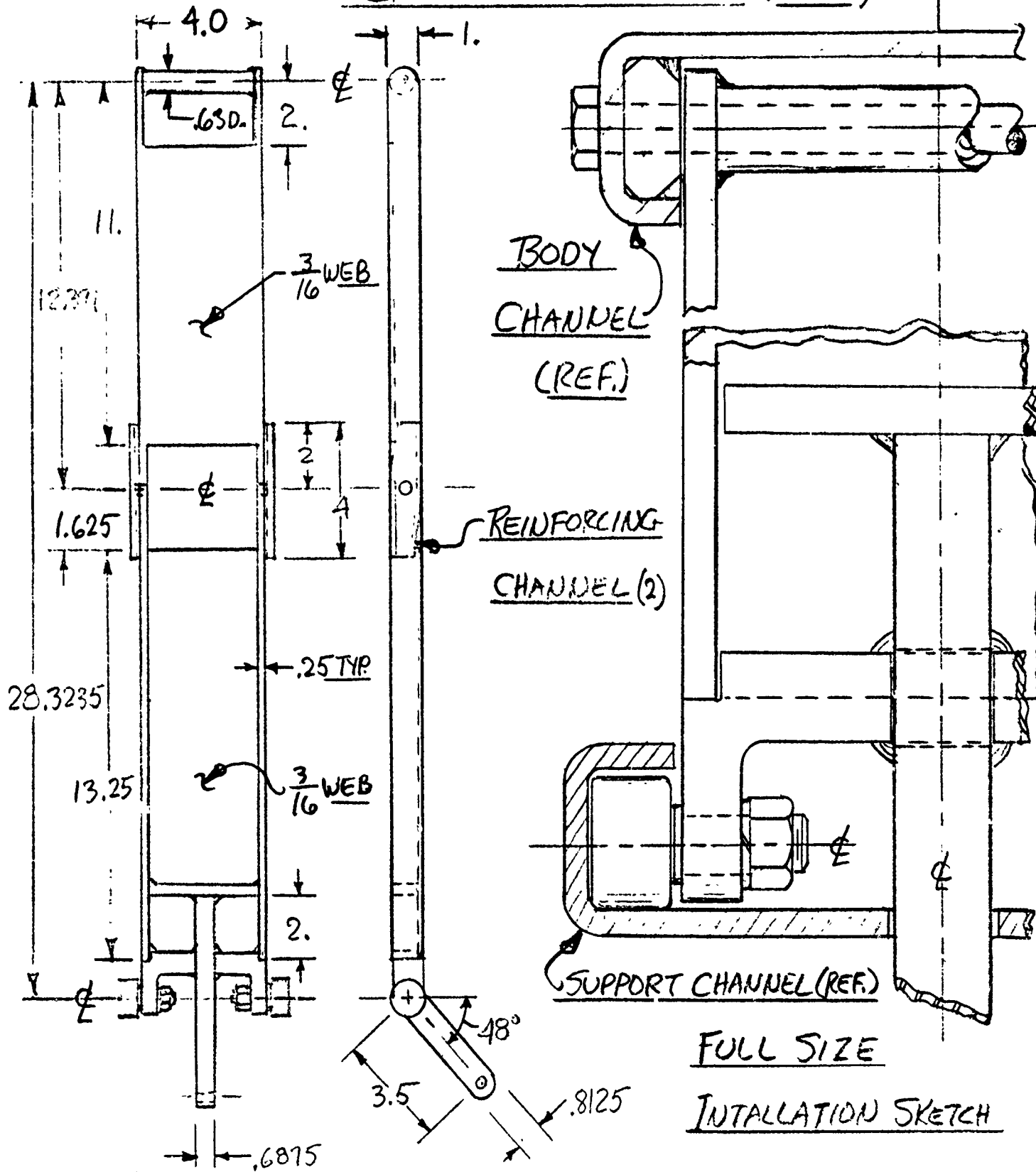
A-L. PAPER

PAGE

65

OF

EJECTOR CHANNEL (No. 1.)



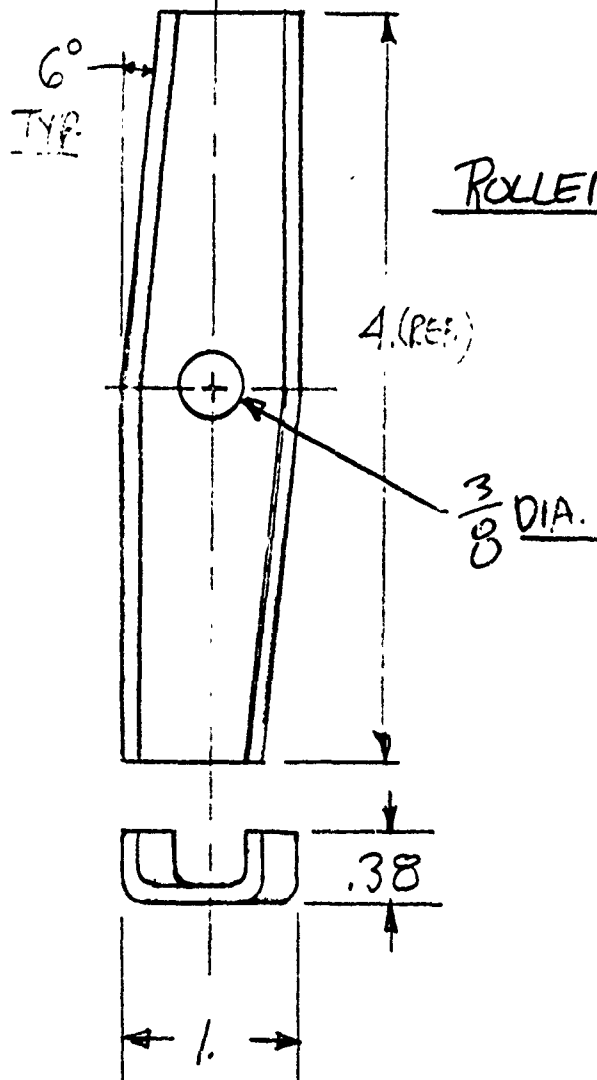
PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME K. Boree
DATE 9-19-79

REFERENCE A-L. RAYNER
PAGE 66 OF

ERECTOR CHANNELS (#1)

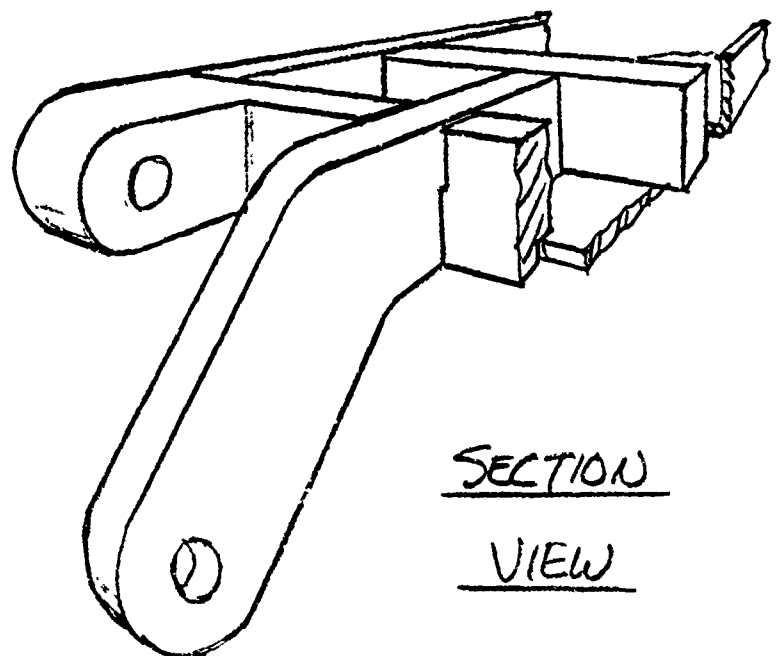
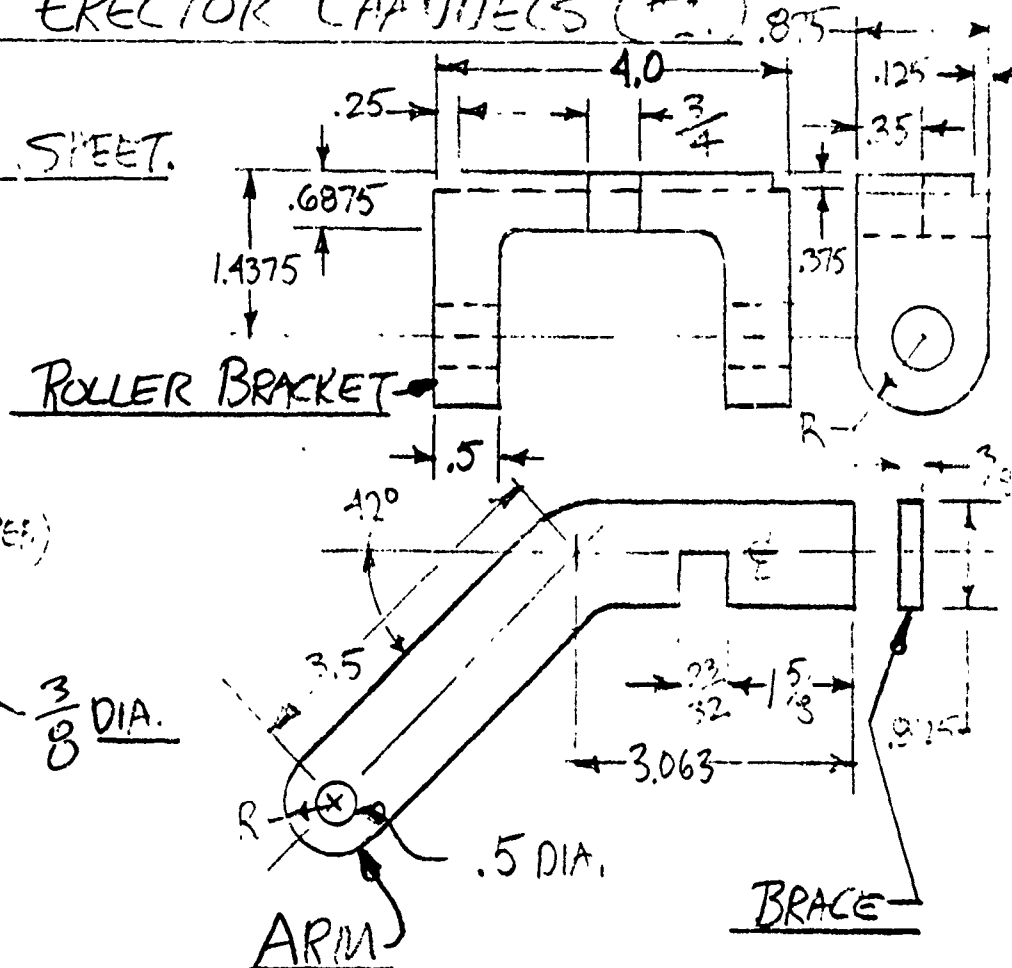
13GA. (0897) STL. SHEET.



REINFORCING
CHANNELS

NOTE: CAN BE MADE
SYMMETRICAL.

PCF-RN-1284



SECTION
VIEW

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME

K. BOUCC

DATE

9-19-79

REFERENCE

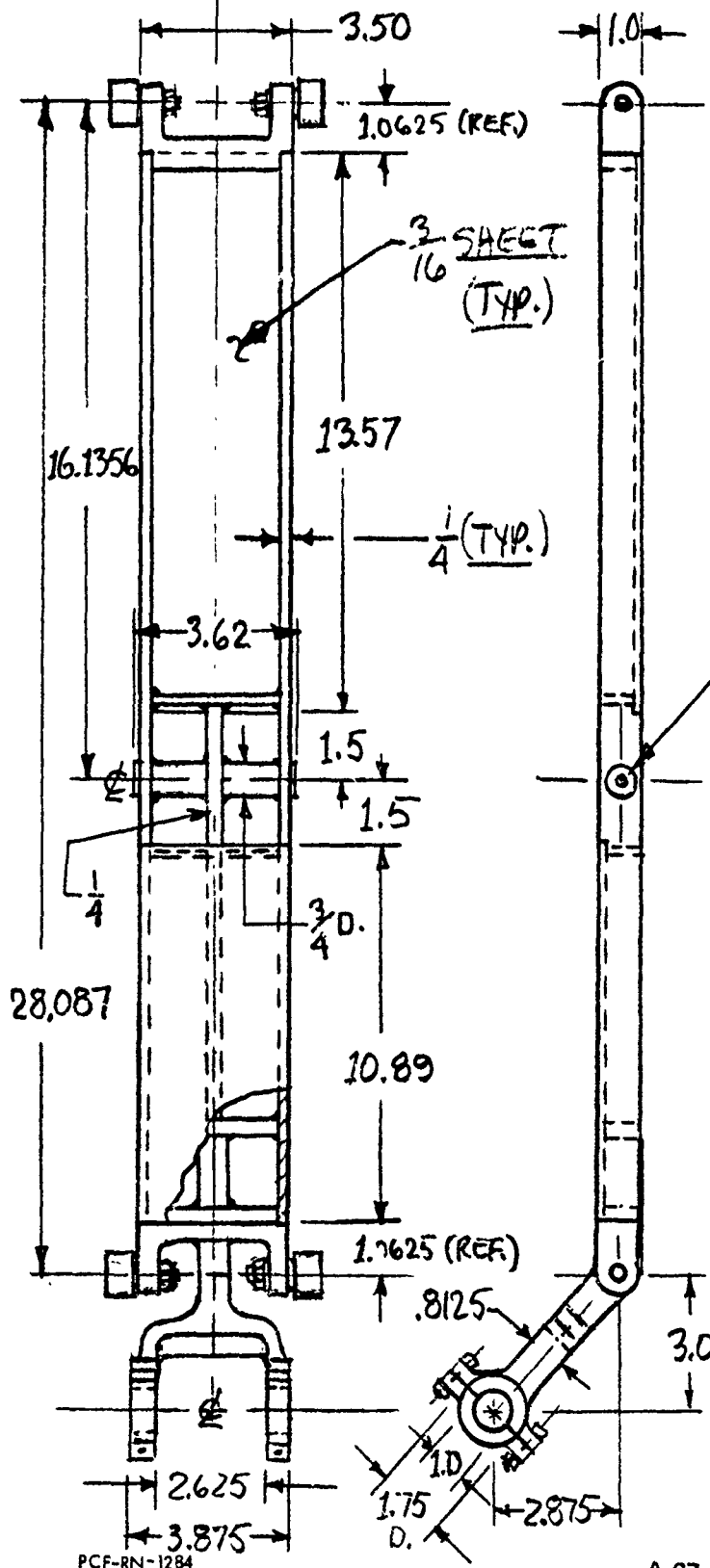
A-L. RAMMER

PAGE

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OF

ERECTOR CHANNELS (No. 2)



NOTES:

1) ROLLERS ON CHANNEL
ENDS ARE - MCGILL CAMROL
BEARINGS. # CF-1-S OR
EQUIVALENT. TWO WASHERS
& A (7/16 NF) NUT REQD, EACH.

HOLE FOR 3/8 BOLT

2) RAMMER ELEVATING
CYLINDER IS - ORTMAN -
MILLER, TRUNDION MOUNTED
CYL., 1500 PSI MAX., SERIES
7L, 1 1/2 IN. BORE, 5/8 IN.
ROD, & A ROD CLEVIS.

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

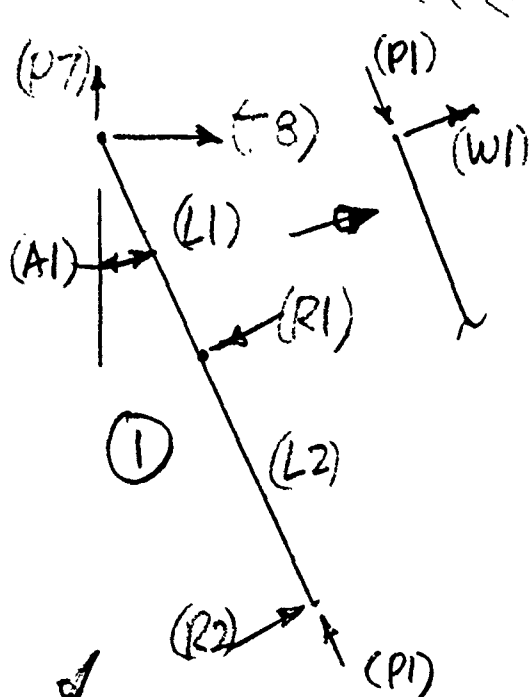
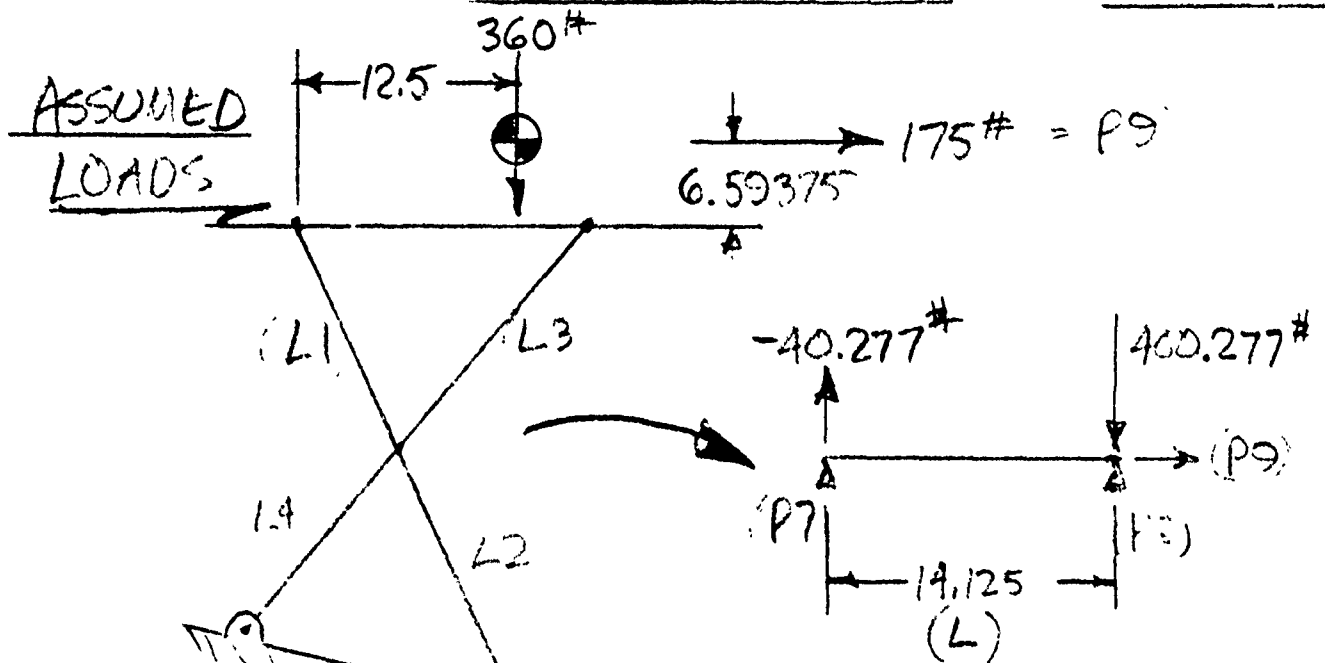
NAME K. BOVEE
DATE 12-11-79

REFERENCE A-L. RANNER
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ERECTOR CHANNELS

STRESS ANALYSIS

CASE 1



$$W1 = T8 \cos(A1) - P7 \sin(A1)$$

$$P1 = T8 \sin(A1) + P7 \cos(A1)$$

$$\sum M = 0;$$

$$W1(L1 + L2) = R1 \cdot L2;$$

$$R1 = \frac{W1(L1 + L2)}{L2}$$

$$\sum V = 0;$$

$$R2 = R1 - W1.$$

No. 1 CHANNEL AS A FREE BODY.

NOTE:

CHARACTERS

ARE 370-B&C

AND CHANNEL

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME

KBouze

REFERENCE

A-L. PAMMER

DATE

12-11-79

PAGE

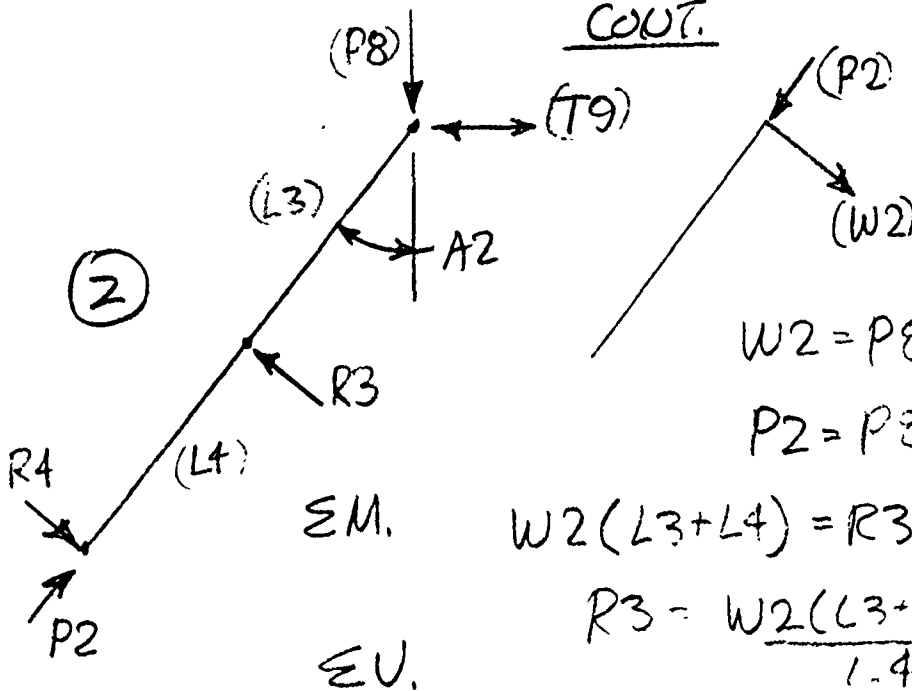
69

OF

ERECTOR CHANNELS

STRESS ANALYSIS

CONT.



$$\underline{T9 = P8 - T8}$$

$$W2 = P8 \sin(A2) + T9 \cos(A2)$$

$$P2 = P8 \cos(A2) - T9 \sin(A2)$$

$$W2(L3 + L4) = R3(L4)$$

$$R3 = \frac{W2(L3 + L4)}{1.4}$$

$$R4 = R3 - W2$$

A COMPUTER PROGRAM, KBDEFL4, (370-BASIC)

WAS WRITTEN TO ANALYZE THE STRUCTURE.

A LOAD (T8) IS ASSUMED, (T9) IS CALCULATED,

& ALL THE OTHER LOADS, ETC., ARE CALCULATED

TO OBTAIN THE TOTAL DEFLECTIONS OF THE TWO

BEAMS. SINCE THE UPPER ENDS OF THE (2)

BEAMS ARE COUPLED, THEIR DEFLECTIONS ARE

RELATIVE: FOR A GIVEN DEFLECTION OF # (1) BEAM

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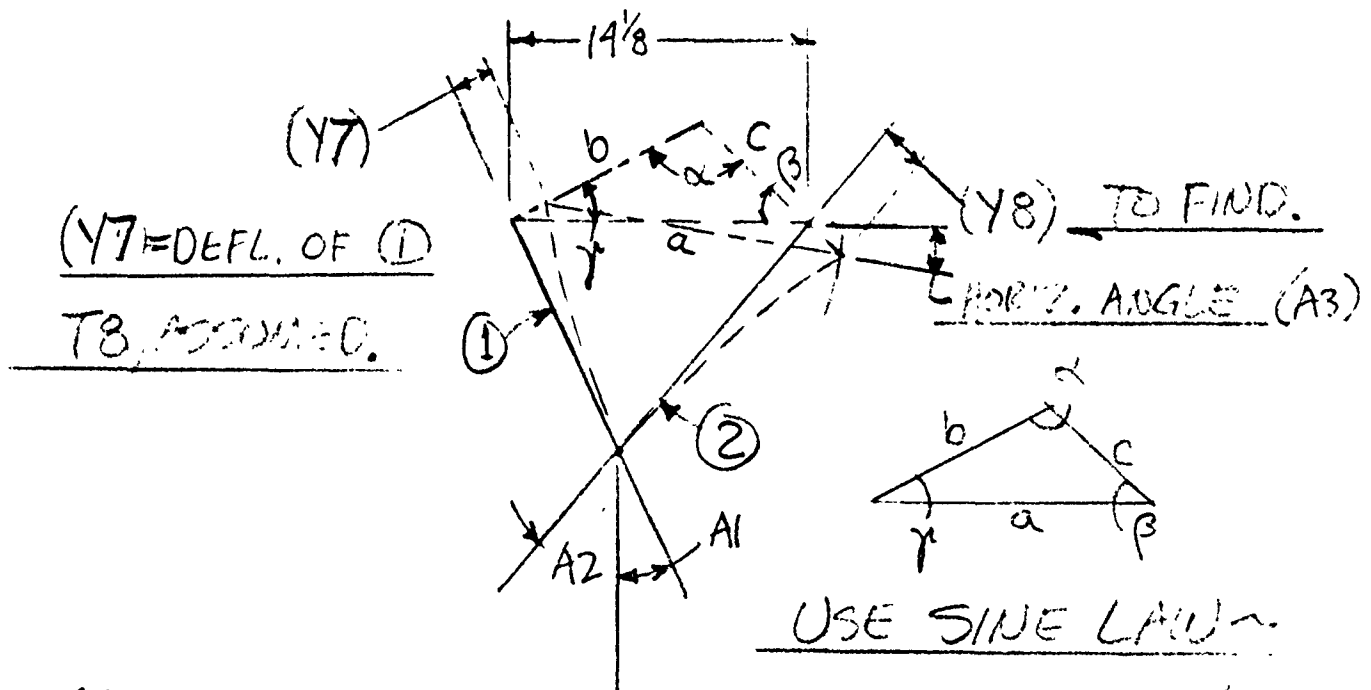
REFERENCE A-L. RAMMER
PAGE 70 OF

ERECTOR CHANNELS

STRESS ANALYSIS

CONT.

- THE DEFLECTION OF # (2) BEAM CAN BE APPROXIMATED AS FOLLOWS:



$$(A1) = .2617993878 \text{ RAD. } (15^\circ) \quad (R) = \frac{a}{\sin \alpha} = \frac{b}{\sin \beta} = \frac{c}{\sin \gamma} = \left(\frac{14.125}{\sin \alpha} \right)$$

$$(A2) = .7417426366 \text{ RAD. } (42.49872256^\circ) \quad a = 14.125 \text{ IN. IS FIXED. } \alpha = 180^\circ - (A1 + A2)$$

$$\gamma = (A1) \neq \beta = (A2) \text{ BEFORE DEFL.}$$

$$\text{SOLVE: } b = \frac{a \sin \beta}{\sin \alpha} \quad (L5)$$

$$c = \frac{a \sin \gamma}{\sin \alpha} \quad (L6)$$

$$a = (L)$$

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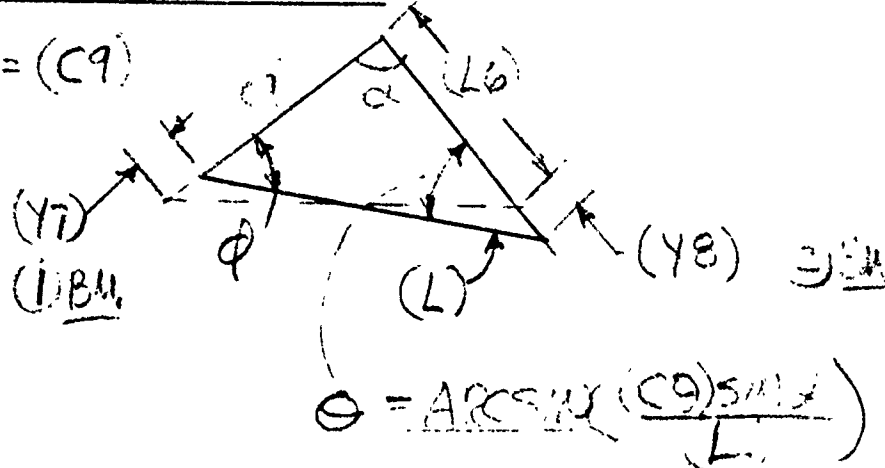
ERECTOR CHANNELS

STRESS ANALYSIS

CONT.

AFTER (Y7) IS CALCULATED:-

$$(L5) - (Y7) = (C9)$$



$$\phi = 180 - \alpha - \theta, \quad (L6) + (Y8) = \frac{a}{\sin \alpha} (\sin \phi) = (R) \sin \alpha$$

(A7) (A9)

$$(Y8) = (A9)(L6) - \text{REQD. DEFLECTION FOR}$$

(2) BEAM TO MATCH.

THE METHOD USED IS TO LOOP THE (T8) LOADS,
& CALCULATE THE (T9) LOADS. THEN CALCULATE
THE DEFLECTIONS OF (1) & (2) BEAMS & FIND A
(Y9) DEFLECTION (OF (2) BEAM) THAT MATCHES
THE (Y8) DEFLECTION CALCULATED ABOVE. IF
DONE, THE (T8) & (T9) LOADS ARE THE RIGHT SIZES.

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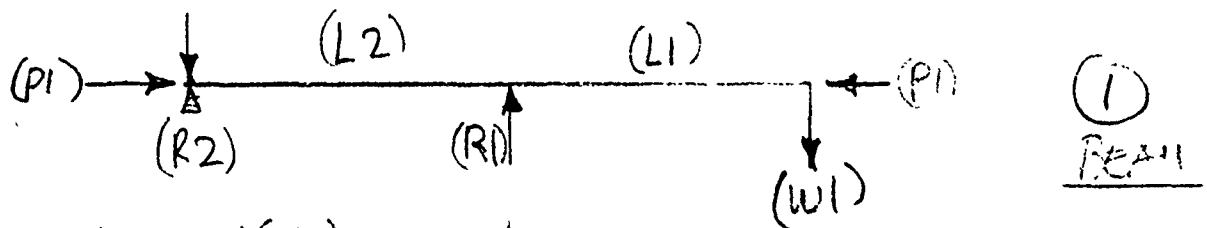
NAME K. B. B. E.
DATE 12-11-79

REFERENCE A-L. RAUWER
PAGE 7? OF

ERECTOR CHANNELS

STRESS ANALYSIS

CONT.

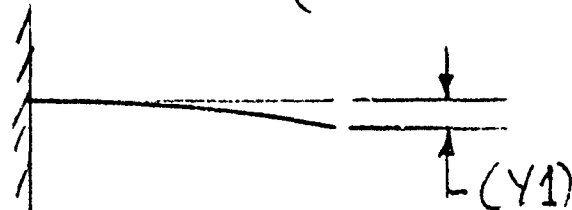


$$M1 = W1 * J1 * \tan(U1)$$

$$Y1 = \frac{W1 * (J1 * \tan(U1) - L1)}{P1}$$

$$J1 = \text{SQR}(E * I1 / P1)$$

$$U1 = L1 / J1$$

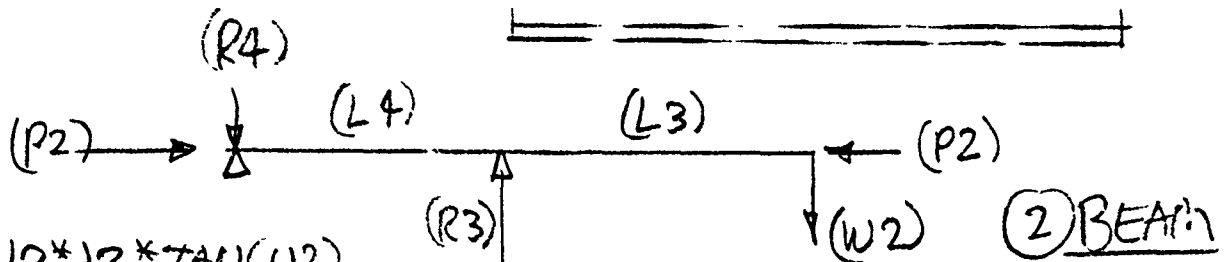


$$M_{max} = -W \tan U$$

$$y = -\frac{W}{P} (J \tan U - L)$$

ROARK FORMULAS:

$$J = \sqrt{\frac{EI}{P}} ; U = \frac{L}{J}$$

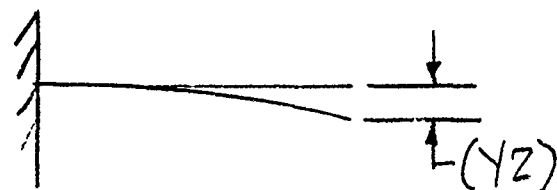


$$M3 = W2 * J2 * \tan(U2)$$

$$Y2 = \frac{W2 * (J2 * \tan(U2) - L3)}{P2}$$

$$J2 = \text{SQR}(E * I2 / P2)$$

$$U2 = L3 / J2$$



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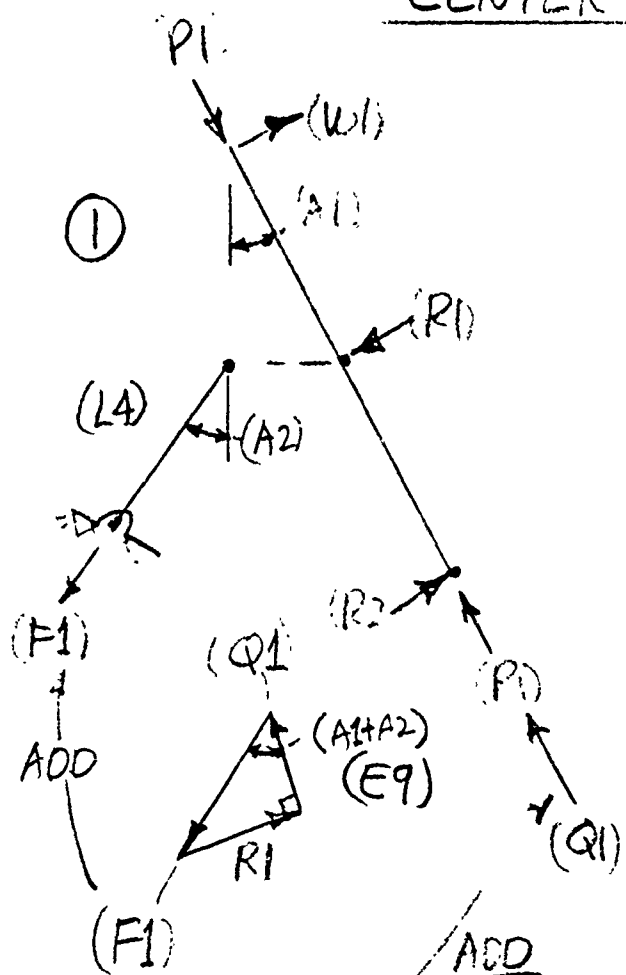
REFERENCE A. L. RAUVER
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ERECTOR CHANNELS

STRESS ANALYSIS

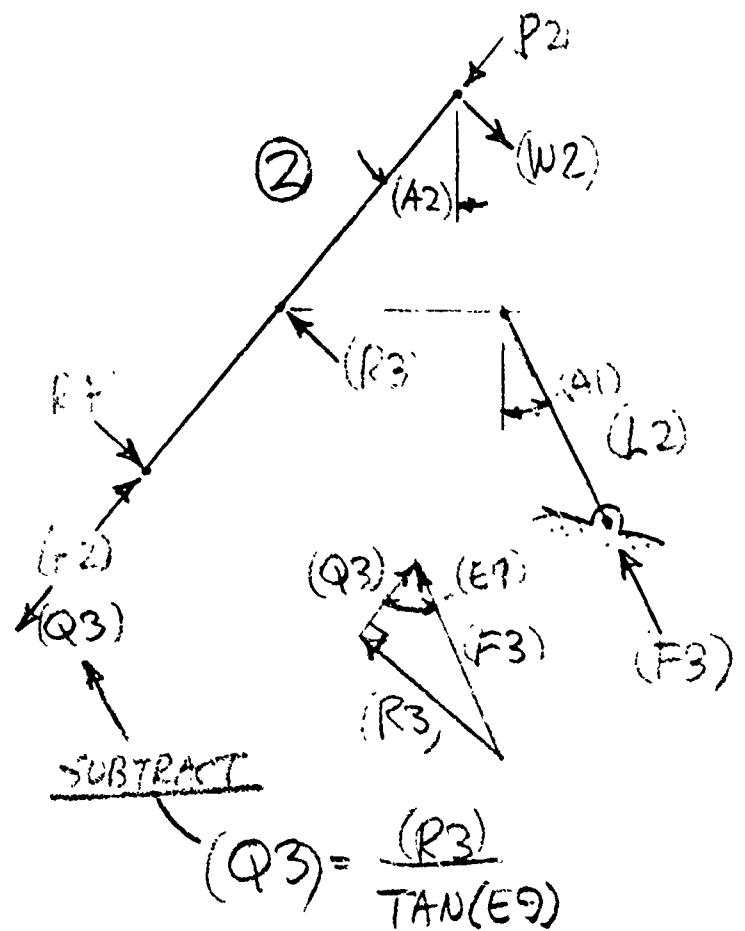
CONT.

CENTER PIN LOADS.



$$(Q1) = \frac{R1}{\tan(E9)}$$

$$(F1) = \text{SQRT}(R1^2 + Q1^2)$$



$$(Q3) = \frac{(R3)}{\tan(E9)}$$

$$(F3) = \text{SQRT}(R3^2 + Q3^2)$$

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DATE 12-11-79

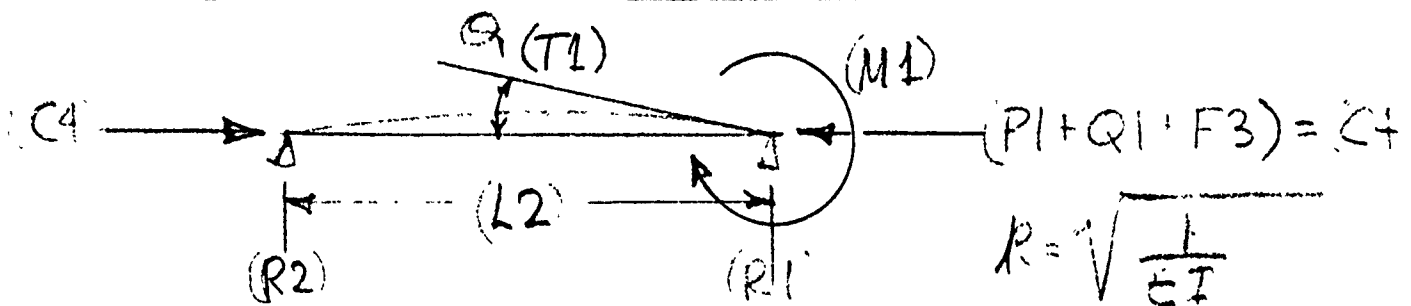
REFERENCE A-L CHAMBER
PAGE 74 OF

ERECTOR CHANNELS

STRESS ANALYSIS

①

CONT.

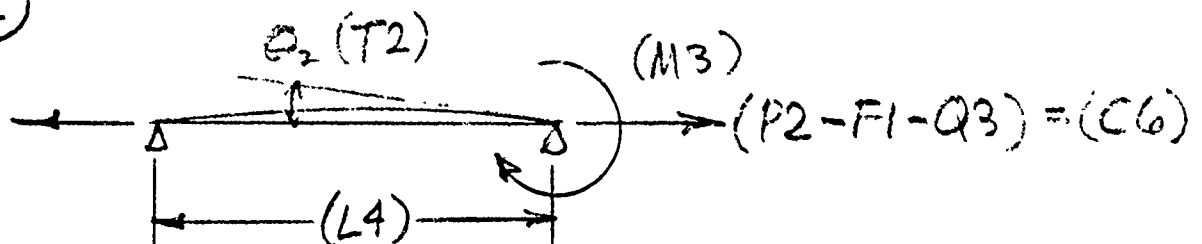


ROARK FORMULA: $\Theta = \frac{-M_0}{Pl} \left(1 - \frac{Rl}{\tan(Rl)} \right)$

$(T1) = M1 * (1 - (J5 * L2 / \tan(J5 * L2))) / (C4 * L2)$

$Y7 = L1 * \sin(T1) + Y1 = \text{TOTAL DEFL. (1)}$

②



ROARK FORMULA: $\Theta = \frac{M_0 K}{P} \left(\frac{1}{kl} - \frac{C_{a1}}{C_2} \right); a=0.$

$(J6) = K;$

$K = \sqrt{\frac{P}{EI}}, C_2 = \sinh kl, C_{a1} = \cosh k(l-a)$

$(T2) = M3 * J6 * (1 / (J6 * L4) - \cosh(J6 * L4) / \sinh(J6 * L4)) / \text{ABS}(C6)$

$(Y9) = L3 * \sin(T2) + Y2 = \text{TOTAL DEFL. (2)}$

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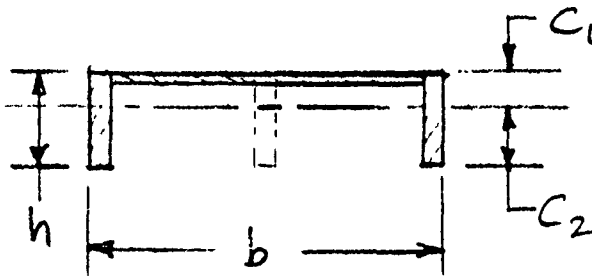
ERECTOR CHANNELS

STRESS ANALYSIS

CONT.

WITH THE LOADS & MOMENTS DETERMINED.

THE STRESSES CAN BE CALCULATED.



TYP. CHANNEL

CROSS-SECTION

A STD. COMPUTER PROGRAM,

KBSECT1, WAS USED TO

DETERMINE THE $I, C_1, & A$

OF EACH CHANNEL.

① L1 ~ $.010425, .269426, 1.15625$

① L2 ~ " " "

② L3 ~ $.087, .284926, 1.0625$

② L4 ~ $.114439, .33149, 1.26502$

STRESS = $\frac{Mc}{I} + \frac{P}{A}$

① L1 ~ $S(1,K) = ABS(M1)/I3$

① L2 ~ $S(2,K) = ABS(M1)/I7 + ABS(C4)/E5$

② L3 ~ $S(3,K) = ABS(M3)/I4$

② L4 ~ $S(4,K) = ABS(M3)/I8 + ABS(C6)/E6$

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

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ERECTOR CHANNELS

STRESS ANALYSIS

CONT.

AFTER TWO SEARCH RUNS, A PRINT-OUT OF
THE FINAL LOADS, DEFLECTIONS, STRESSES, ETC., WAS
OBTAINED.

250 T8=257.822
500 REM.
700 REM.
RUN

KBDEFL4 06:44 12/12/79 WEDNESDAY 106

PROG. CALCS. DEFL., LOADS, MOMS. & STRESSES OF RAMMER ERECTOR CHANS.

NO.	#1 HOR. LD.	#1 DEFL.	#2 DEFL. REQ.
1	257.822*	-.142738 in.	-.185753 in.

MAX. BEND. STRESSES IN CHANNELS.

NO.	#1-T	#1-B	#2-T
1	9551.3 PSI.	10332.5 PSI.	11228.4 PSI.

#2 HOR. LD.	#2 DEFL.	HOR. ANGLE
-82.822*	-.18575 in.	0.98833°

#2-B
10422.2 PSI.

$$\frac{F_{ty}}{F_{act}} = \frac{36000}{11228.4} = 3.2 \text{ F.S. OF SAFETY.}$$

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME KBOWEE
DATE 12-19-79

REFERENCE A-L 3411ER
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ERECTOR CHANNELS

STRESS ANALYSIS

CONT.

CASE 2.

CASE 1 CONSIDERED ALL THE JOINTS AS PIN JOINTS.

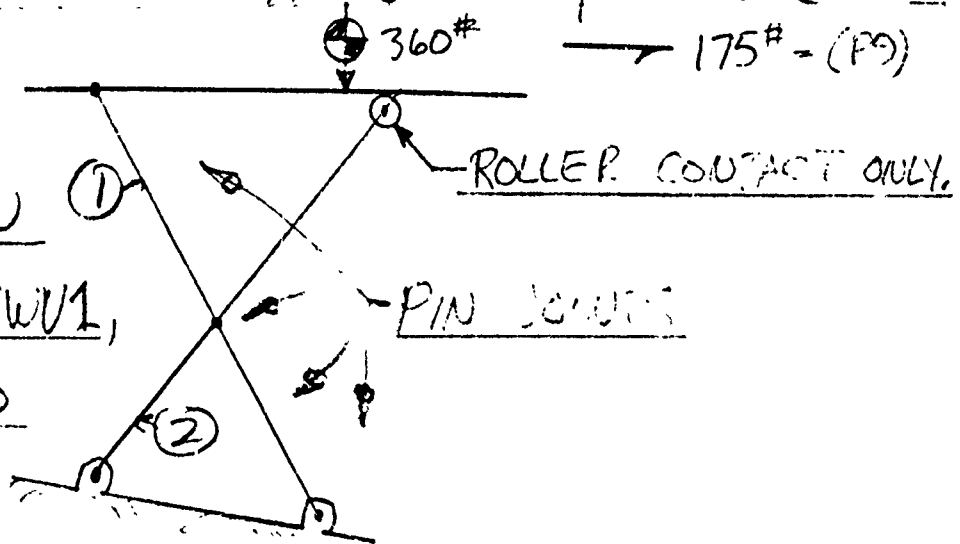
CASE 2 WILL CONSIDER ALL JOINTS FIXED & ROLLER CONTACT ONLY.

A WEAVE & RUN

PROGRAM, KBDEFWV1,

WAS WRITTEN TO

SOLVE THIS



CASE. ALL THE HORIZONTAL LOAD IS CARRIED BY ①

CHANNEL. THEN A DIFFERENT GEOMETRY WAS USED

TO FIND THE DEFLECTION ANGLE OF THE BODY CHANNEL.

(A3)

KBDEFWV1 12:48 12/19/79 WEDNESDAY 106

250 T8=P9

572 X4=Y9#COS(A2)

574 Y4=Y9#SIN(A2)

575 L9=L+X4-X3

576 A3=DEG(ATN((Y3+Y4)/L9))

580 REM

760 REM

1090 X3=Y7#COS(A1)

1095 Y3=Y7#SIN(A1)

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME K. Bovee
DATE 12-19-79

REFERENCE A-L. Bovee
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ERECTOR CHANNELS

STRESS ANALYSIS

CONT.

WVR KBDEFL4,KBDEFWV1

12:49 12/19/79 WEDNESDAY .106

PROG. CALCS. DEFL., LOADS, MOMS. & STRESSES OF RAMMER ERECTOR CHANS.

NO.	#1 HOR. LD.	#1 DEFL.	#2 DEFL. REQ.
1	175	-9.86428E-02	-.128643

MAX. BEND. STRESSES IN CHANNELS.

NO.	#1-T	#1-B	#2-T
1	6603.6	7434.82	14475.

#2 HOR. LD.	#2 DEFL.	HOR. ANGLE
0	-.239408	.755235

#2-B

13303

NOTE THAT THE ACTUAL #2 DEFL. IS MORE THAN THE
REQD. #2 DEFL. BECAUSE THE JOINT IS NOT PINNED

$$\frac{F_{en}}{Fact} = \frac{36000}{14475} = 2.487 \text{ FACTOR OF SAFETY.}$$

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME

K. BOE

REFERENCE

A-L. FAYMEC

DATE

12-12-79

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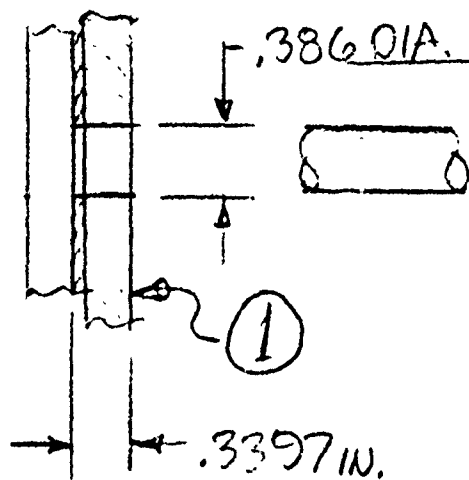
OF

ERECTOR CHANNELS

STRESS ANALYSIS

CONT.

BEARING LOAD CHECK.



3/8 DIA. STL. (D2)

MAX. LOAD (C4) = 905 LBS.

USE ROARK FORMULA -

$$S_{c \max} = 0.591 \sqrt{pE \frac{D_1 - D_2}{D_1 D_2}}$$

$$p = \frac{905(1.15)}{2(.3397)} \text{ lb/in.} = 1532 \text{ #/in.}$$

$$S_{c \max} = 0.591 \sqrt{1532(29 \text{ mil}) \frac{.386 - .375}{.386(.375)}} = 34,340.2 \text{ PSI}$$

$$F_{bo} = \frac{90000}{34340.2} = 2.62 \text{ FACTOR OF SAFETY.}$$

NOTE: ABOVE CASE HAS THE MIN. BEARING AREA.

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME

Conner

REFERENCE

A-L-2-4-1-R

DATE

12-12-79

PAGE

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OF

ERECTOR CHANNELS

STRESS ANALYSIS

CONT.

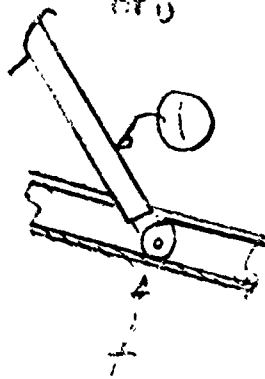
BEARING STRESS UNDER CHILLER BEARINGS.

USE HOBBS FORMULA: $S_e = 3190 \sqrt{\frac{P}{D}}$

$P = \frac{374(1.15)}{2(.5)} = 350 \frac{\text{lb}}{\text{in.}}, D = 1" \quad (1.15 = \text{ASSUMED DYNAMIC FACTOR})$

$S_e = 3190 \sqrt{\frac{350}{1}} = 59,679.44 \text{ PSI} *$

$F_{Br0} = \frac{90000}{59679.44} = 1.503 \text{ FACTOR OF SAFETY}$



MAX. CASE.

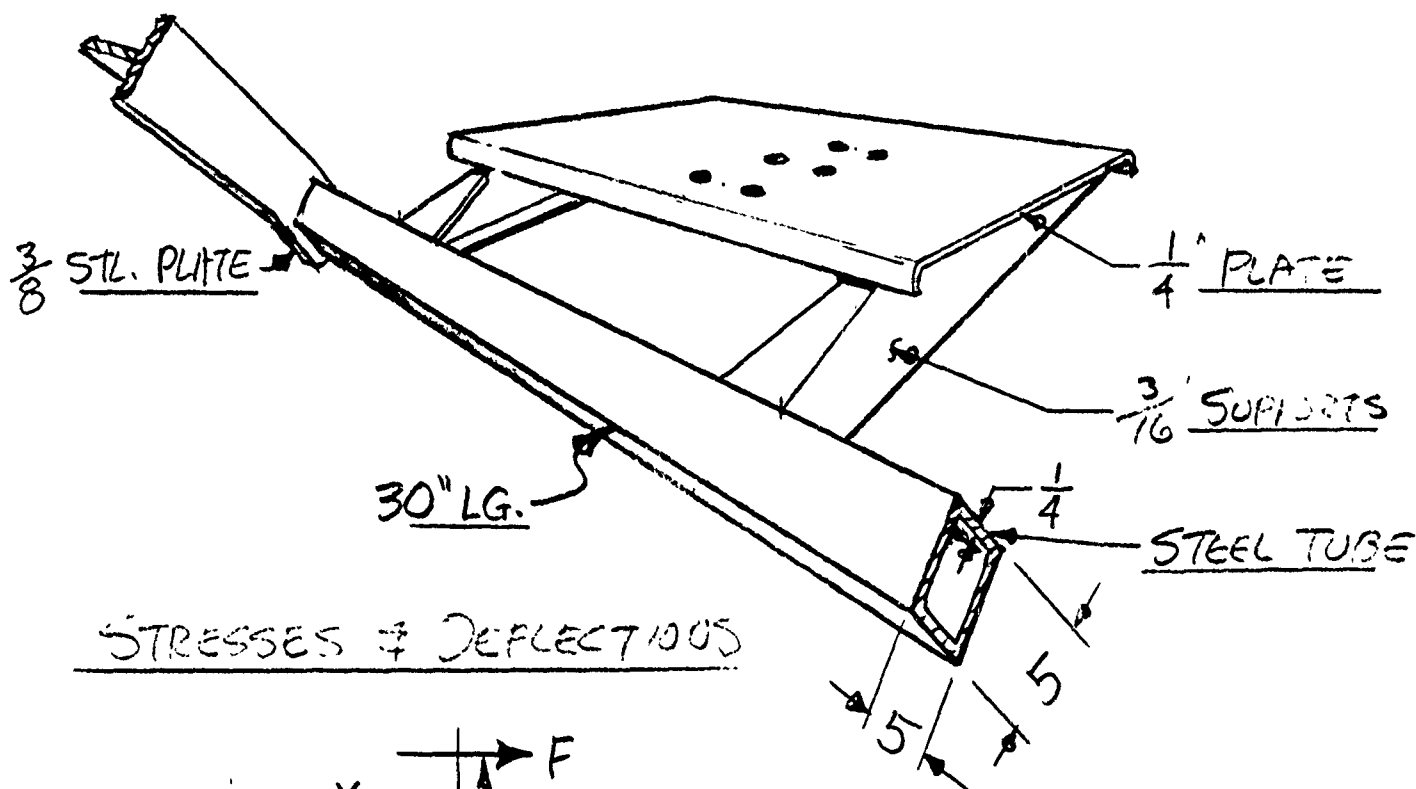
PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME KBOUZE
DATE 9-6-79

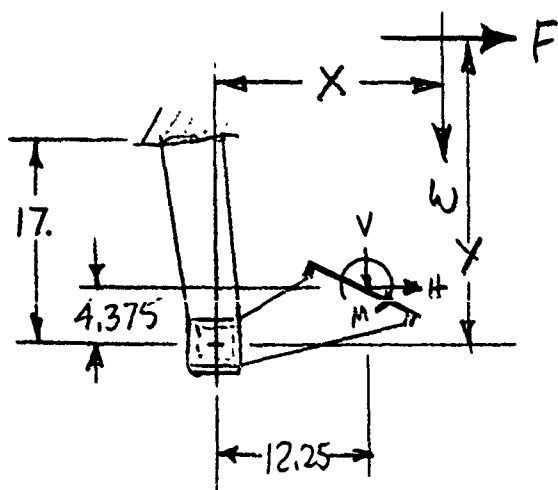
REFERENCE A-L RAMMER
PAGE 81 OF

SUPPORT, RAMMER ASSY.

THE SUPPORT CHANNEL IS BOLTED TO THE $\frac{1}{4}$ "
PLATE SHOWN BELOW WITH (6) $\frac{3}{8}$ BOLTS. HYDRAULIC
CONTROL ITEMS CAN BE ALSO MOUNTED ON THIS PLATE



STRESSES & DEFLECTIONS



ESTIMATES: F = 177# (x 2)

W = 250#

X = 28"

Y = 35.5"

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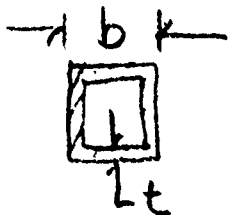
REFERENCE A-L RAMMER
PAGE 82 OF

SUPPORT-RAMMER ASSY.

STRESSES & DEFLECTIONS (CONT.)

$$MOM._{TOT} = 177(2)35.5 + 250(28) = 19567 \text{ #"}^2$$

(NOTE: NO INERTIA FORCES HAVE BEEN CALCULATED.)



$$R = t(b-t)^3 = .25(5-.25)^3 = 26.793$$

$$f = \frac{T}{2t(b-t)^2} = \frac{19567}{2(.25)(5-.25)^2} = 1734.471 \text{ psi}$$

$$\frac{35000(.6)}{1734.5} = 12.1 \text{ F.S.}$$

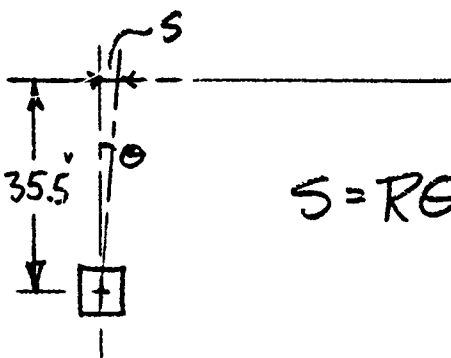
$$\theta = \frac{TL}{GR} = \frac{19567(15)}{1150000(2)26.793} = 0.0004763 \text{ RAD.}$$

$$= 0.0273^\circ$$

$$= 0^\circ - 1' - 38.29''$$

$$S = R\theta = 35.5(.0004763) = 0.01690865'$$

MOVEMENT.



DEFLECTIONS DUE TO INTERMEDIATE

STRUCTURAL MEMBERS WILL ADD TO THIS FIGURE &

THE INERTIA FORCES WILL SUBTRACT.

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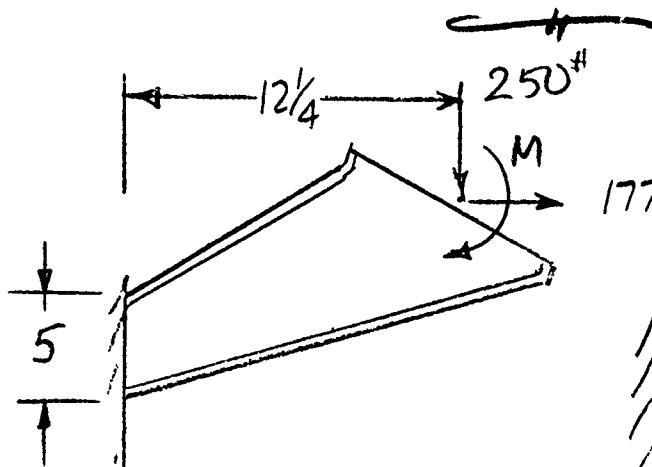
NAME K. B. ROE
DATE 9-7-79

REFERENCE A-L RAMMER
PAGE 83 OF

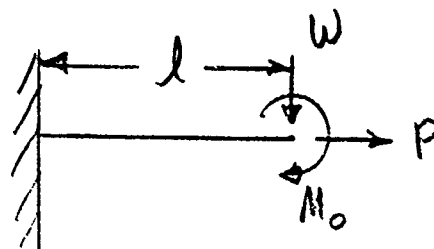
SUPPORT-RAMMER ASSY.

DEFLECTIONS (CONT.)

DEFLECTIONS OF THIS MAGNITUDE CAN BE EASILY COMPENSATED FOR BY ADJUSTING THE POSITION & ATTITUDE OF THE BODY CHANNEL. THIS IS DONE BY JUDICIOUS PLACEMENT OF THE TWO (2) ADJUSTABLE ROLLER-TRAVEL-STOPS, ONE IN THE BODY CHANNEL & ONE IN THE SUPPORT CHANNEL. IN ANY EVENT, THE OPTIMUM "AIM" OF THE RAMMER 'GUN' WILL HAVE TO BE DETERMINED BY ACTUAL OPERATION & TEST. SUFFICIENT ADJUSTMENT MEANS HAS BEEN PROVIDED.



$$M = 177(2) 31.125 + 250(15.75) = 14,955.75^{\#}$$



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DATE 9-7-79

REFERENCE A-L. RAMMER
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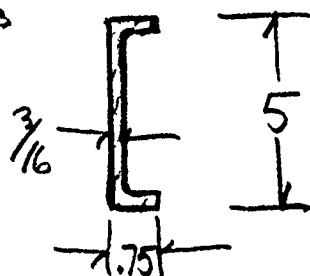
SUPPORT-RAMMER ASSY.

USING ROARK:

$$J = \sqrt{\frac{EI}{P}}; U = \frac{l}{J}$$

$$\text{MAX. } M = -WJ \tanh U, y_{\text{max}} = \frac{W}{P}(l - J \tanh U)$$

$$I = \frac{BH^3 - bh^3}{12}$$



$$W = 250 \text{ \#}$$

$$P = 354 \text{ \#}$$

$$l = 12.25 \text{ \#}$$

$$I = \frac{.75(5^3) - .5625(4.625^3)}{12} = 3.175 \text{ IN}^4$$

$$J = 509.9988 / \text{use } 510. \quad U = .02401961$$

$$M_{\text{max}} = -250(510) \tanh(.02402) = 3061.96 \text{ \#"}.$$

$$y_{\text{max}} = \frac{250}{354}(12.25 - 510 \tanh(.02402)) = .0015222 \text{ '}$$

NOTE: ONLY ONE CHANNEL X-SECT. WAS USED,
HENCE M & y ARE TWICE ACTUAL FIGS.

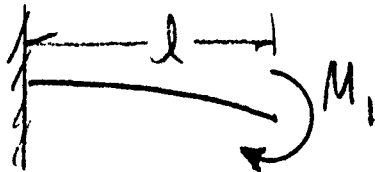
$$\text{ADD } M_0 \text{ \& } P(4\frac{3}{8}) \text{ TO } M_{\text{max}} = 16501.5 + 3061.96 \\ = 19566.46 \text{ \#"} \text{ TOTAL.}$$

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REFERENCE A-L RAMMER
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SUPPORT-RAMMER ASSY.



$$y_1 = \frac{M_1 l^2}{2EI} = \frac{16504.5 (12.25)^2}{2900000 (3.175)}$$

$$y_1 = .0268187948 \text{ "}$$

$$y_m + y_1 = .028421 \text{ " *}$$

* THIS WOULD BE CORRECT FOR A UNIFORM BEAM
A FACTOR OF SAFETY OF (2). HOWEVER, THE
SECTION TAKEN WAS THE SMALLEST # THE
DEFLECTION CALCULATED IS MEANINGLESS.

CALC. STRESSES & WELD STRENGTHS.

$$F_t = \frac{M}{I/c} = \frac{19566.46 (2.5)}{3.175} = 15,406.66 \text{ PSI}$$

$$\frac{36000}{15406.66} = 2.3367 \text{ F.S.}$$

WELD STRGTH.

$$S_w = .75(5) + \frac{5^2}{3}; f_b = \frac{M}{S_w} = \frac{19566.46 (6)}{12.08333} = 9715.76$$

$$A_w = 5 \quad f_s = \frac{W}{A_w} = \frac{250}{5} = 50.0$$

NAME

K. BOVEE

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A-L RAMMER

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OF

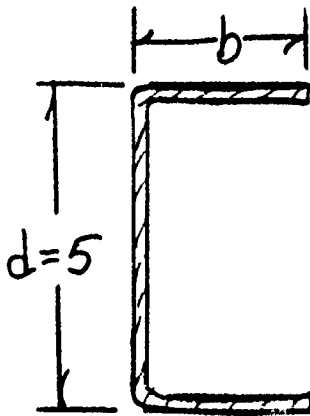
SUPPORT - RAMMER ASSY.

WELD STRENGTH (CONT.)

$$f_r = \sqrt{f_b^2 + f_s^2} = 9715.89 \text{ #/in. (@ F.S. = 2)}$$

E60 WELD ROD \rightarrow $\frac{3}{16}$ FILLET WELD $\approx 1800 \text{ #/in.}$

RECALC. S_w .



INCREASE "b". ★

$$S_w = bd + \frac{d^2}{6}$$

$$f_r = \sqrt{f_b^2 + 50^2} = \frac{1800}{1.5}$$

$$f_b^2 + 50^2 = 1200^2; f_b = 1198.957821$$

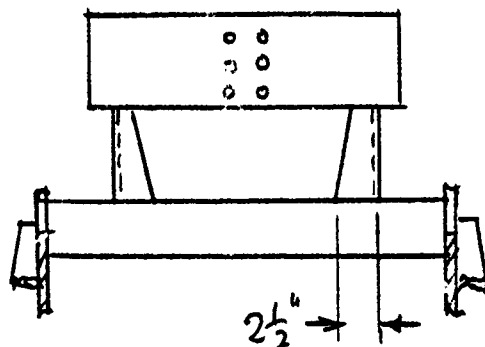
$$f_b = \frac{M}{S_w} = \frac{19566.5}{S_w} = 1199.; S_w = 16.319$$

$$16.32 = 5b + \frac{25}{6}; b = 2.430667"; \text{USE } \underline{\underline{2.5"}} \text{ ✓}$$

CHECK.

$$f_r = \sqrt{1199^2 + 50^2} = 1200 \text{ #/in}$$

$$\frac{1800}{1200} = 1.5 \text{ F.S.}$$



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OF

BOX DETAILS.

THE MAGAZINE BOXES ARE DESIGNED TO BE
REMOVABLE. THE BASIC STRUCTURE OF THE BOXES
IS IDENTICAL: THEY ARE MADE INTO L & R HAND
BOXES BY MACHINING THE SPIRAL RELIEF
GROOVES IN THE SHELF EDGES ADJACENT
TO THE ELEVATOR SECTION. THEN THE FLAP
DOORS ARE ADDED TO THIS SAME SIDE, & THE
BEARINGS FOR THE HORIZONTAL SPIRAL
DRIVE RODS ARE WELDED TO THE OTHER
(OUTER) SIDE.

THE BOXES ARE MADE FROM $\frac{1}{16}$, $\frac{1}{8}$ & $\frac{3}{16}$ MILD STEEL
SHEET, CHANNELS, & ANGLES, IN A WELDED ASSEMBLY.
THE HORIZONTAL SPIRAL RELIEF GROOVES ARE
MADE OF NYLON, TEFLON, ETC., BLOCKS CEMENTED
TO THE METAL SHELF STRUCTURE.

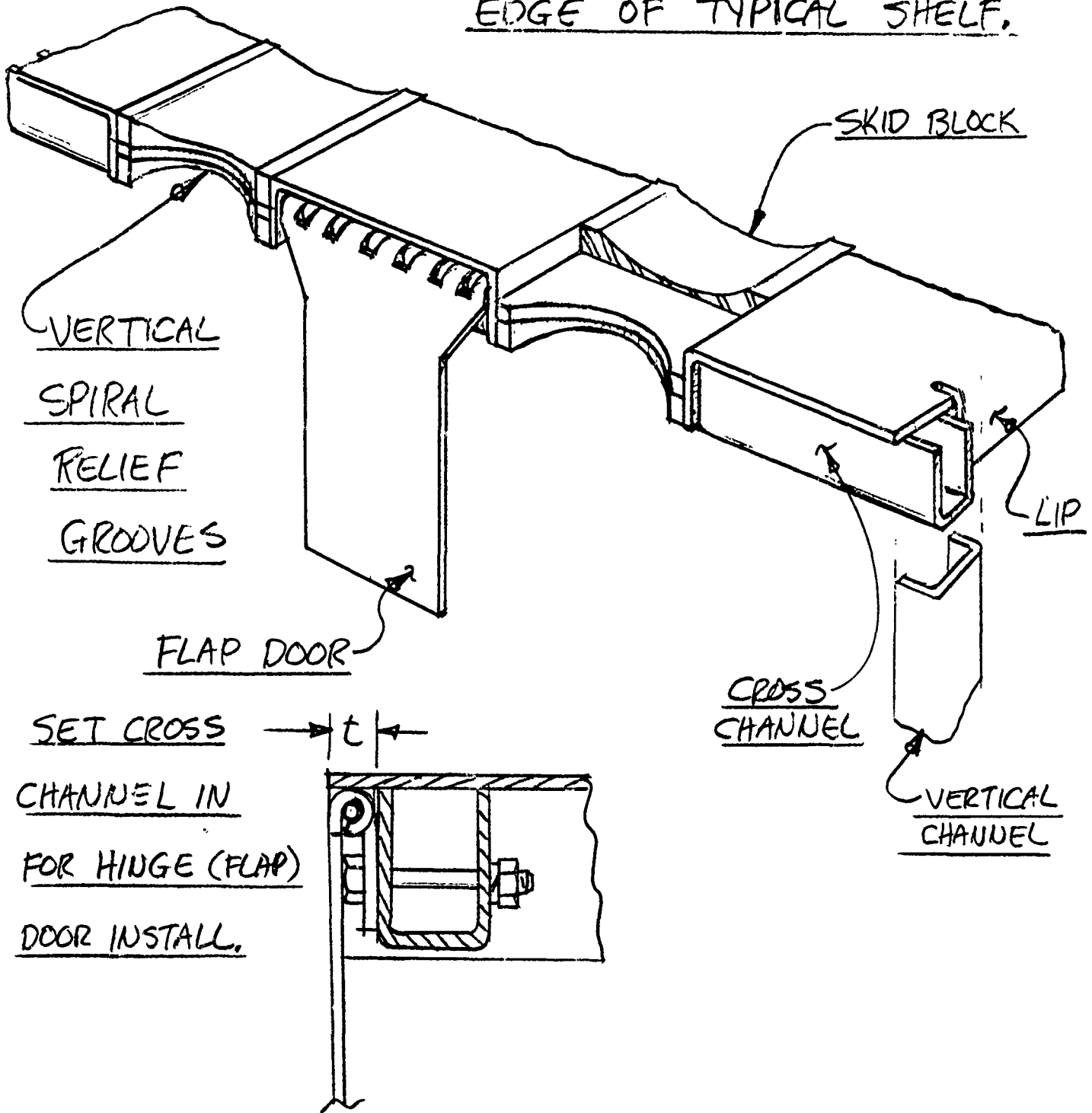
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BOX DETAILS

PARTIAL SECTION VIEW OF INNER
EDGE OF TYPICAL SHELF.



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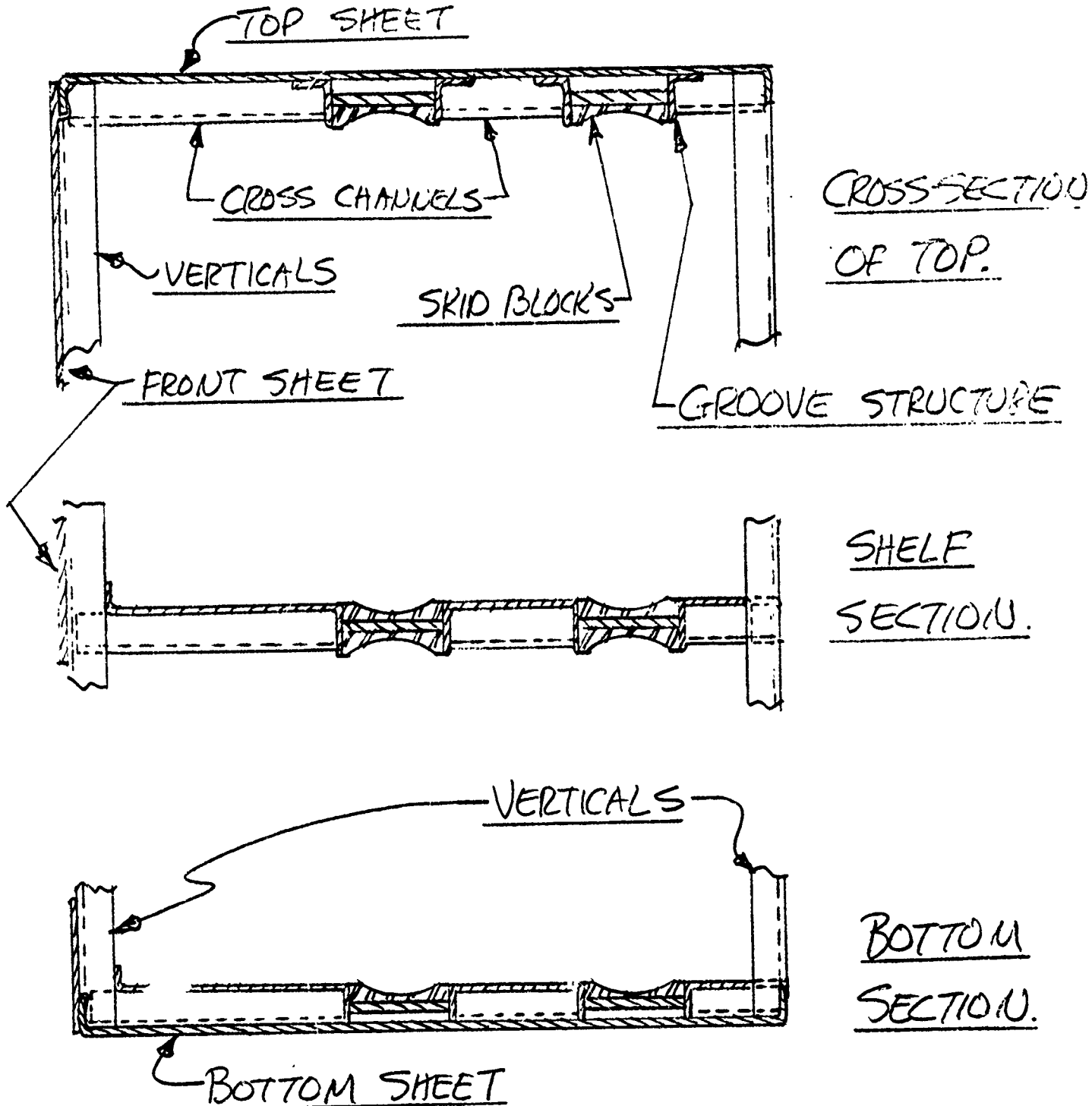
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BOX DETAILS



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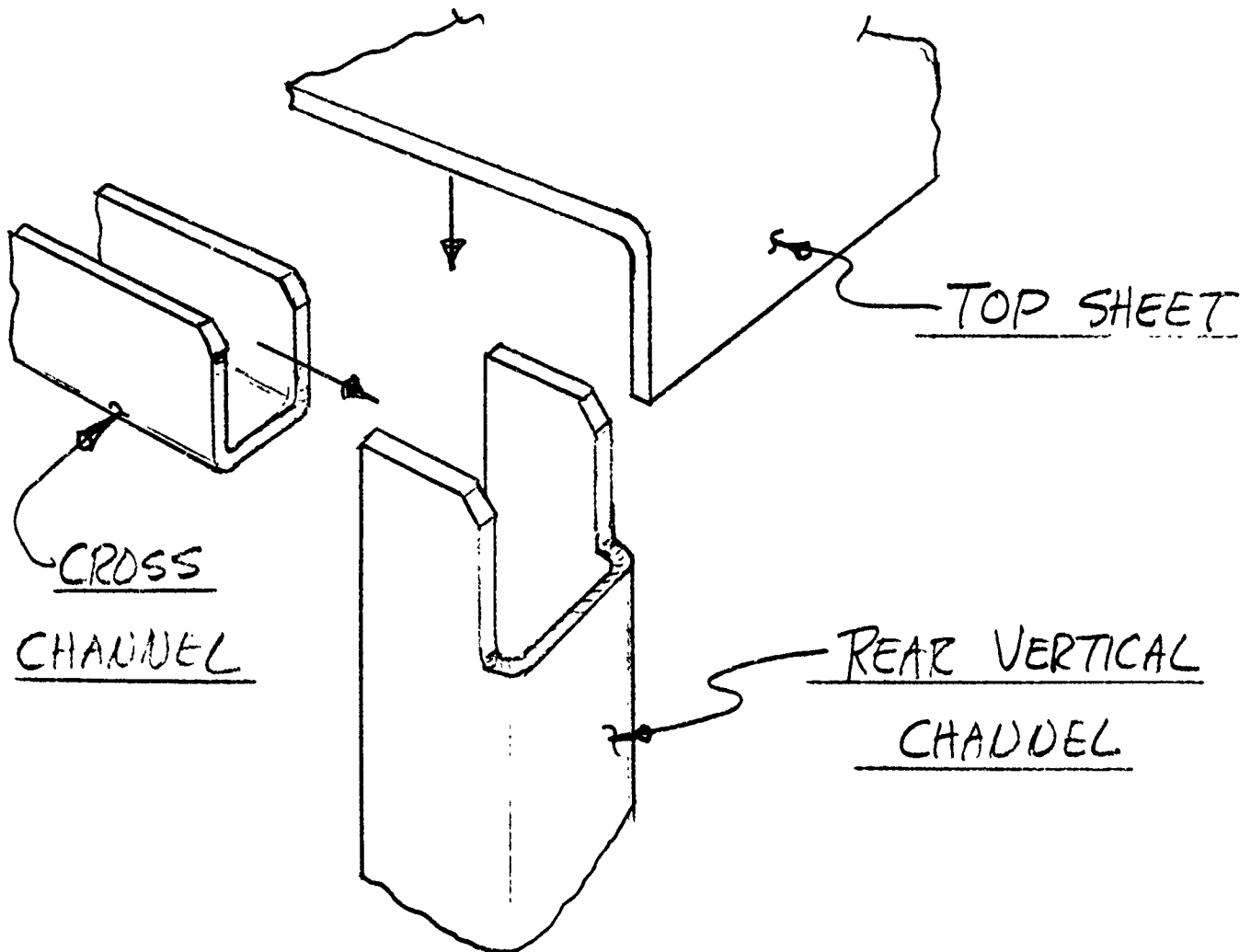
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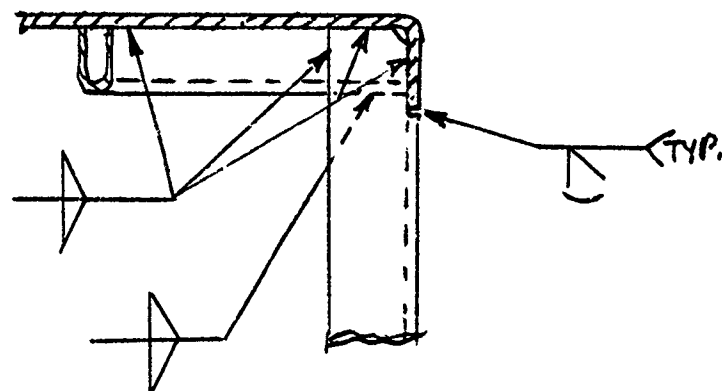
OF

BOX DETAILS



TYPICAL TOP EDGE
CONSTRUCTION.

NOTE: BOTTOM IS
SIMILAR - CROSS
CHANNEL W/B REVERSED.



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OF

SUPPORT BASE

THE BASE STRUCTURE CAN BE UTILIZED
NOT ONLY FOR STOWAGE - ROUNDS, POWDER
CANNISTERS, ETC., BUT FOR THE INSTALLATION
OF MOST OF THE HYDRAULIC & ELECTRONIC
EQUIPMENT & GEAR NECESSARY FOR THE AUTO-
LOADER CONTROL & OPERATION. THE ONLY 'OUT-OF-
BASE' HYDRAULICS ARE ON THE RAMMER SYSTEM.
WERE THE ABOVE ACCOMPLISHED, ADAPTATION
OF THE DESIGN FOR KIT MANUFACTURE, STORAGE,
& INSTALLATION WOULD BE GREATLY FACILITATED.

THE REAR TRACK SUPPORTS & THE CARRIAGE
OPERATING SYSTEM IS ALSO FASTENED TO THE
BASE, EVEN THO NOT INDICATED IN THE SKETCH.
THESE ITEMS APPEAR ON SHEET 1 OF THE DWG.

—+—

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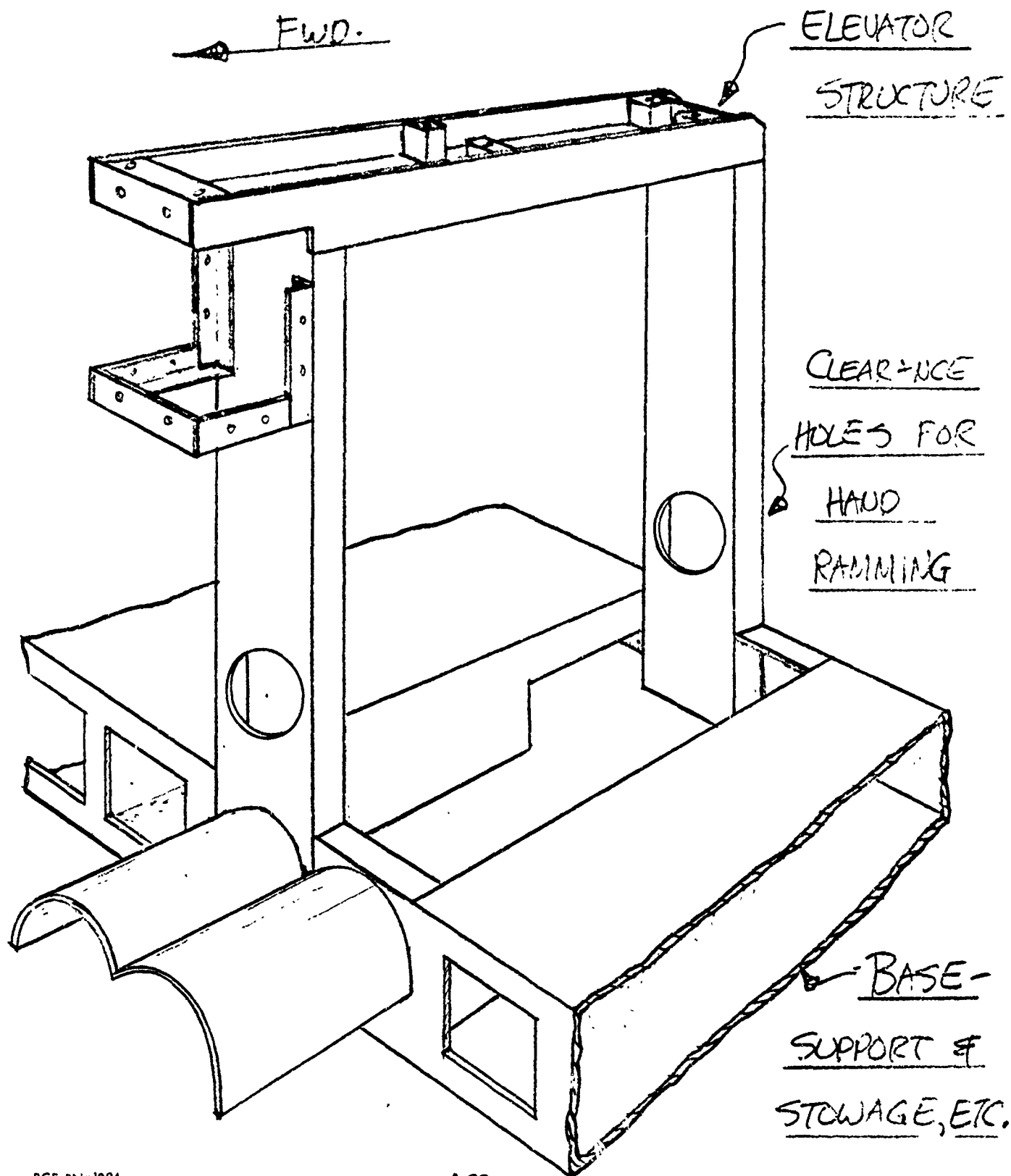
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OF

CENTER STRUCTURE



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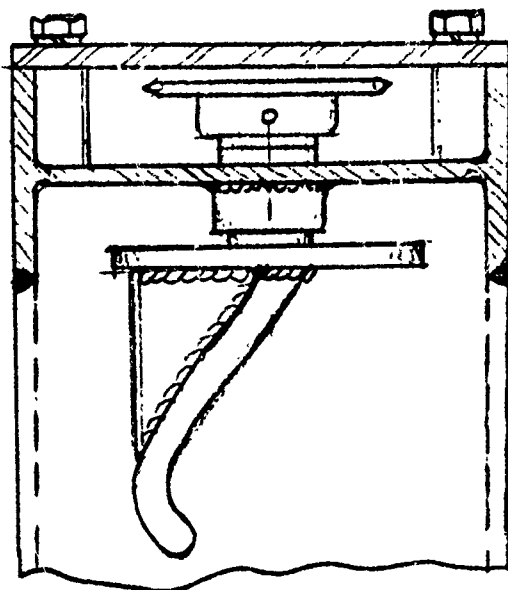
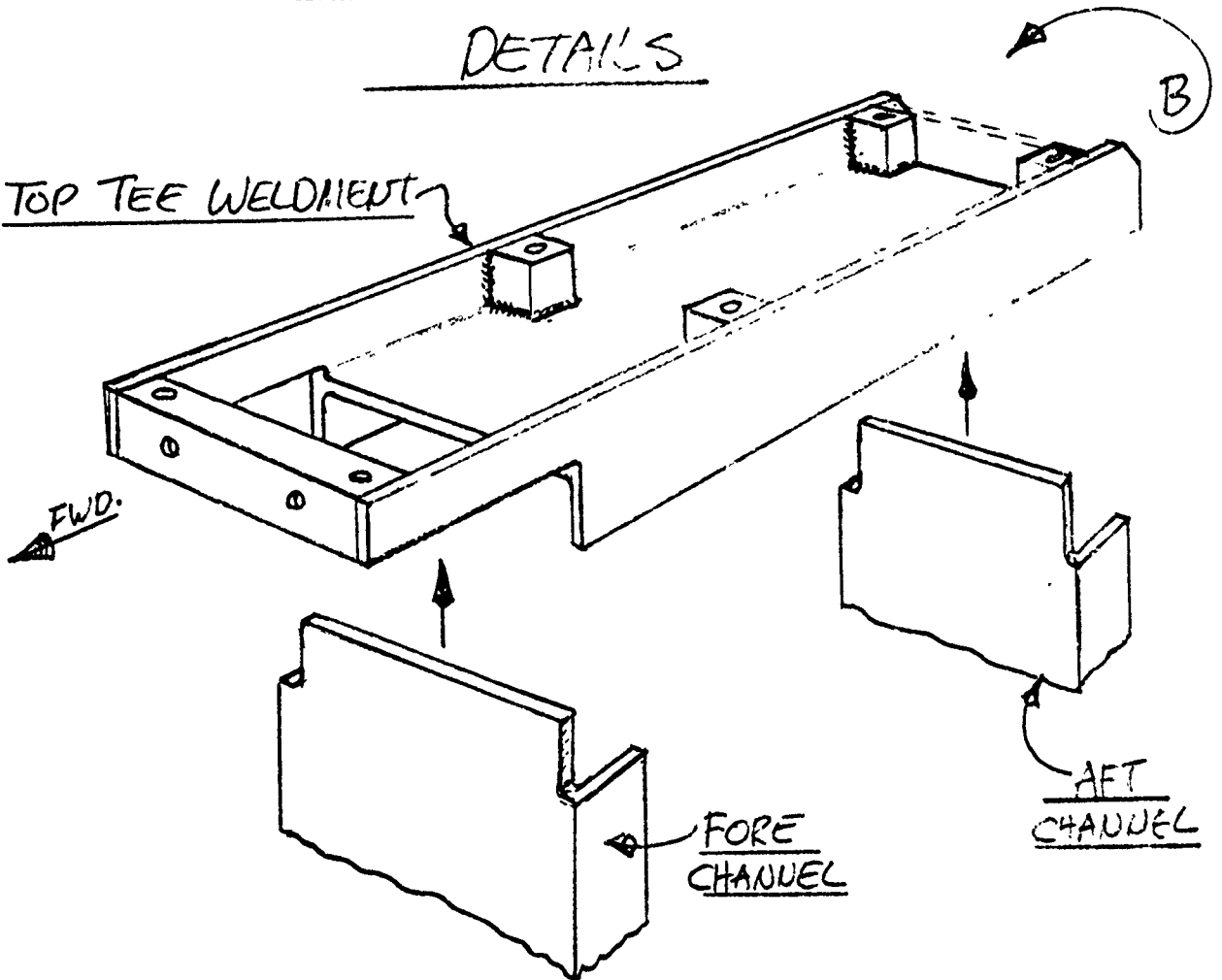
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OF

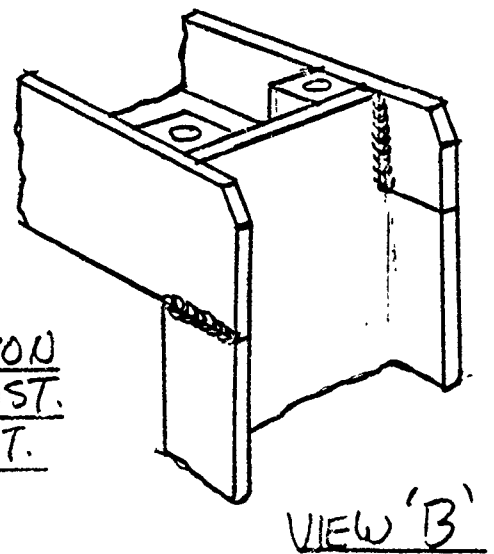
CENTER STRUCTURE

DETAILS

TOP TEE WELDMENT



CROSS-SECTION
OF TEE INST.
LOOKING AFT.



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OF

SPIRAL ROD GENERATION.

THE IDEA OF USING A SPIRAL ROD (SCREW)
IN A CONVEYING SYSTEM IS NOT NEW.
BUT IN THIS CASE IT WOULD HAVE THE AD-
VANTAGES OF SIMPLICITY AND ONE-WAY MOTION.
NO PART OF IT HAS TO RETURN UNLOADED - AS
IN A BELT CONVEYOR SYSTEM.

TO DETERMINE THE DIMENSIONS OF THE
SPIRAL ROD (SPRING) A SEARCH PROGRAM WAS
WRITTEN GENERATING SINE WAVES THAT
WOULD BE TANGENT TO, OR LARGER THAN, A
BASE CIRCLE (ROUND DIA. + A CLEARANCE = $\frac{1}{2}$ ROD
DIA.) SUBJECT TO CERTAIN LIMITATIONS. THE
MOST PROMISING WAVES WERE STUDIED BY
WEAVING & RUNNING THE SEARCH PROGRAM AND
A 'PRINT-OUT' PROGRAM TOGETHER. THE FINAL
SELECTION PRINT-OUT & LISTING IS ON THE NEXT
SHEET. ALL SPIRAL RODS IN THIS DESIGN ARE
THE SAME.

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OF

SPIRAL ROD GENERATION.

PROGRAM PRINT-OUT FOR CURVE X-Y COORDS.

12:34 02/27/80 WEDNESDAY 106

X	Y	L	R	ANGLE (E)
0	0	4.02768	4.02768	0
.402768	.567075	3.62491	3.669	8
.805537	1.12019	3.22215	3.41131	19
1.2083	1.64571	2.81938	3.26455	30
1.61107	2.13072	2.41661	3.2218	41
2.01384	2.56326	2.01384	3.25973	51
2.41661	2.93268	1.61107	3.34607	61
2.81938	3.2299	1.20831	3.44851	69
3.22215	3.44758	.805538	3.54044	76
3.62491	3.58037	.40277	3.60295	83
4.02768	3.625	1.90735E-06	3.625	89
4.43045	3.58037	.402766	3.60295	83
4.83322	3.44758	.805534	3.54044	76
5.23599	3.2299	1.2083	3.44851	69
5.63875	2.93269	1.61107	3.34607	61
6.04152	2.56327	2.01384	3.25974	51
6.44429	2.13073	2.41661	3.2218	41
6.84706	1.64572	2.81938	3.26455	30
7.24983	1.12019	3.22214	3.41131	19
7.65259	.567081	3.62491	3.669	8
8.05536	5.76975E-06	4.02768	4.02768	0
A= 3.625	B= .39			

PROCESSING

1 UNITS

IN.

IN.

DEG

MIN.

SECS.

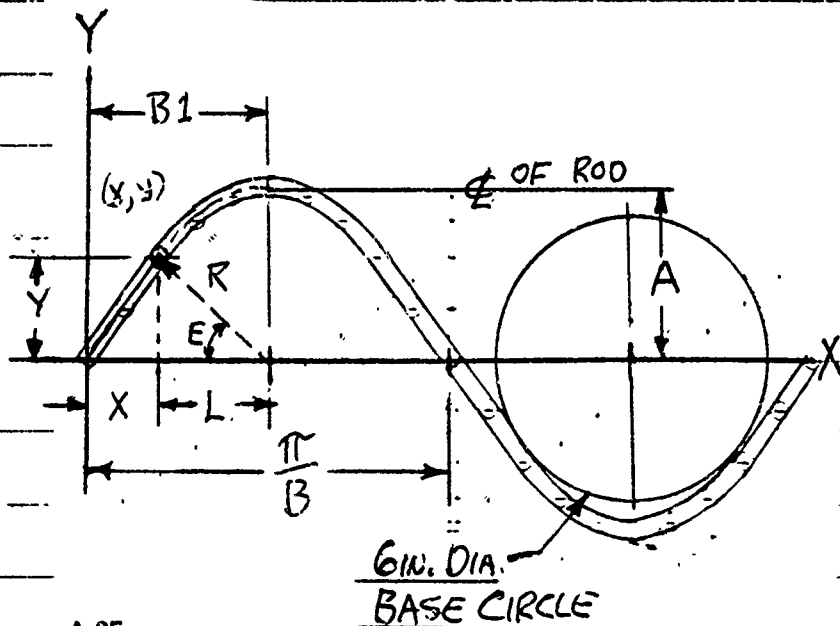
LIST

12:36 02/27/80 WEDNESDAY 106

```

10 A=3.625
20 B=.39
25 PRINT 'X','Y','L','R','ANGLE'
30 B1=6PI/(2*B)
40 B2=B1/10
45 B3=2*B1
50 FOR X=0 TO B3 STEP B2
60 Y=A*SIN(B*X)
65 IF X>B1 THEN 74
70 L=B1-X
72 GO TO 80
74 L=X-B1
80 R=SQR(L^2+Y^2)
82 REM
84 REM
85 IF L=0 THEN 90
86 E=90
88 GO TO 100
90 E=DEG(ATN(Y/L))
92 E1=INT(E)
94 E4=(E-E1)*60
96 E2=INT(E4)
98 E3=(E4-E2)*60
100 PRINT X,Y,L,R,E1,E2,E3
110 IF CPU>9 THEN 160
120 NEXT X
130 REM
140 PRINT 'A=';A,'B=';B
150 GO TO 170
160 PRINT 'CPU>9.'
170 END

```



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A-L. MAGAZINE

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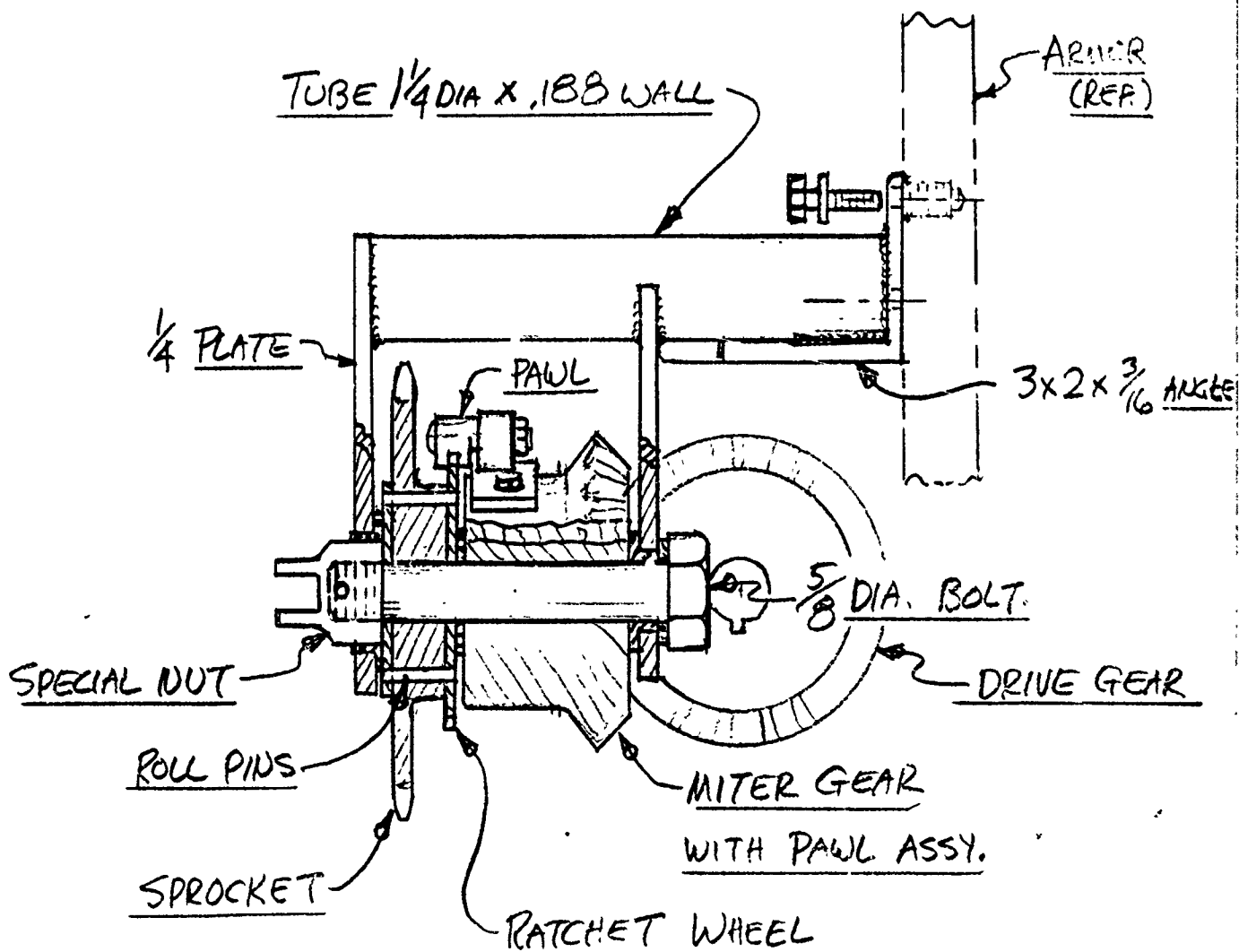
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OF

SPIRAL CONVEYOR DRIVES

THE SPIRAL ROD CONVEYOR MUST ROTATE
180° TO ADVANCE A ROUND ONE SPACE. AT
LEAST THREE (3) SCHEMES CAN BE USED
TO PROVIDE THE POWER & CONTROL. THE GENERAL
MECHANISM SKETCHED BELOW CAN BE USED
FOR ALL THREE SCHEMES WITH MINOR ADAPTATIONS.

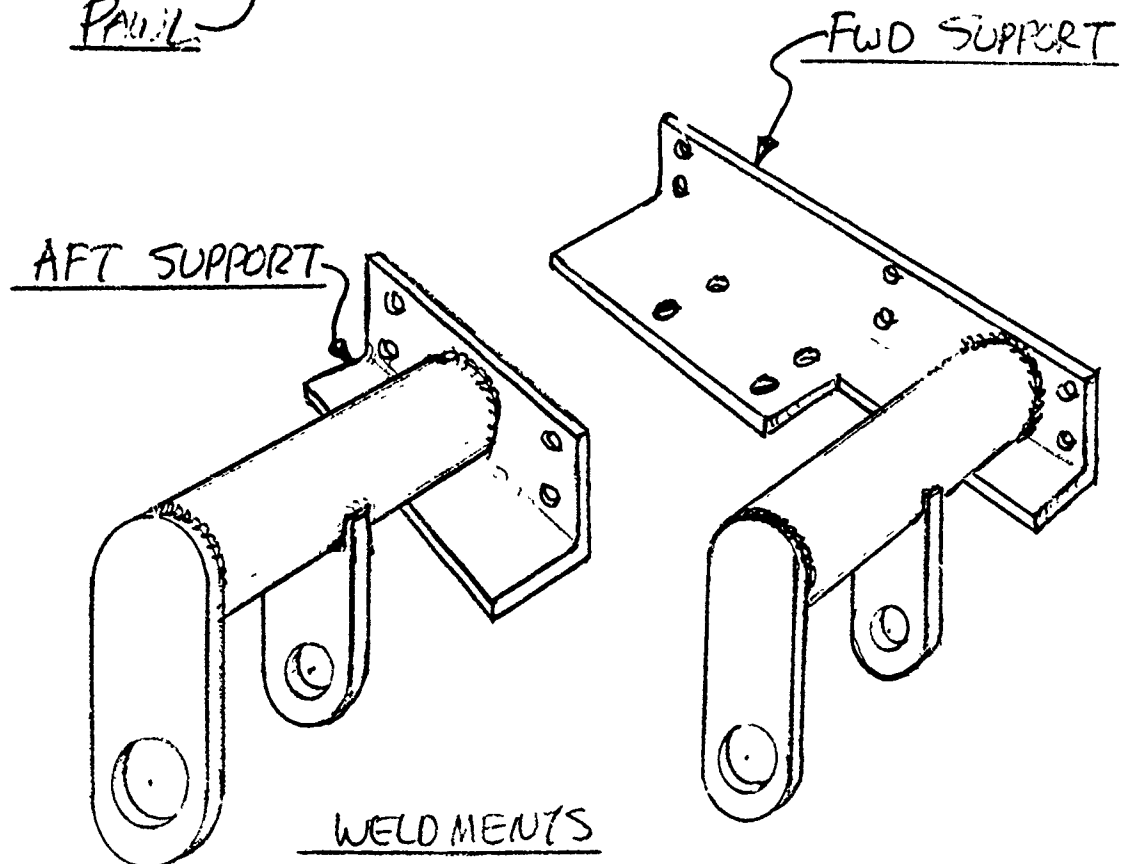
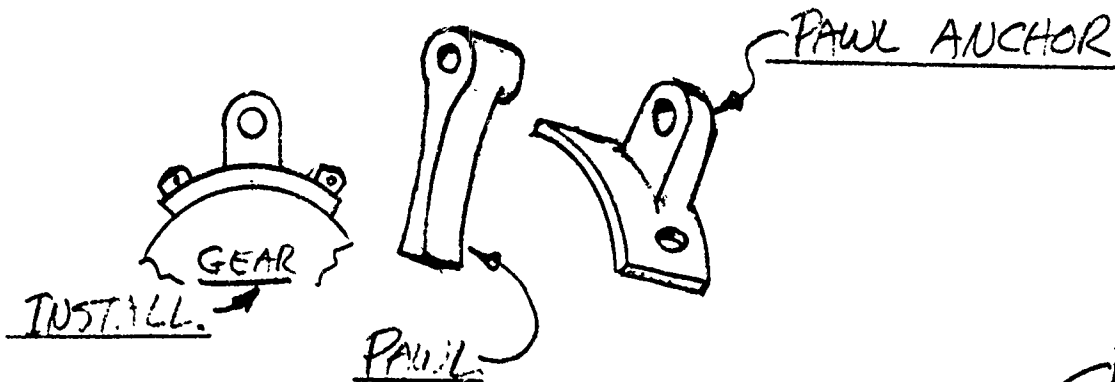
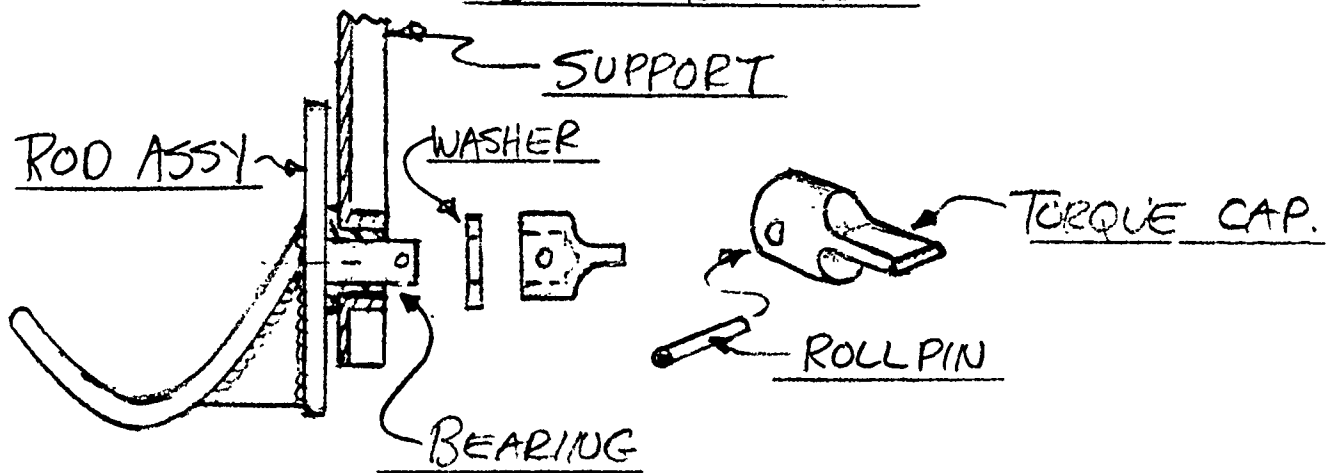


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DRIVE DETAILS



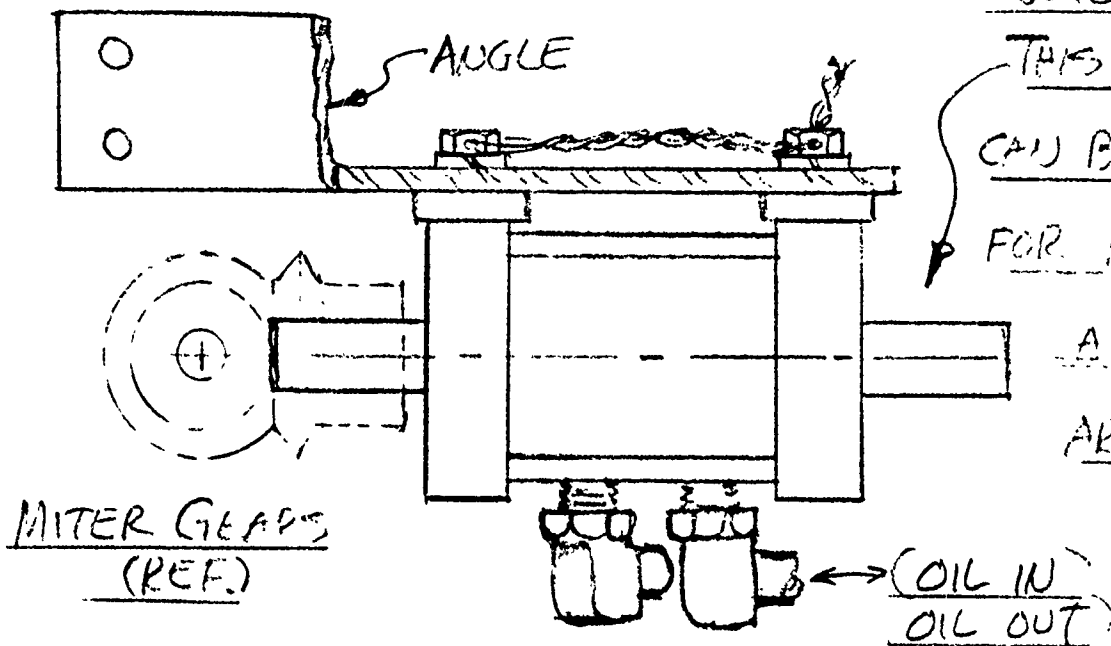
WELDMENTS

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DRIVE DETAILS



TORK-MOR ROTARY ACTUATORS.

360 IN. LB. TORQUE @ 300 PSI 100° ± 5°

MODEL NO. SF-2-2

OR

1400 IN. LB. TORQUE @ 500 PSI 30° ± 5°

MODEL NO. DS-3-3

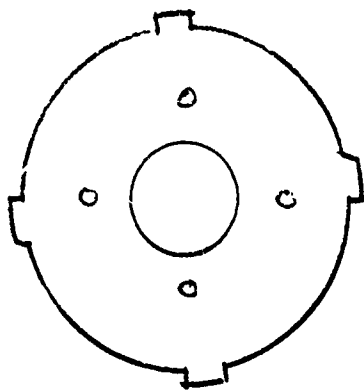
OR

700 IN. LB. TORQUE @ 500 PSI, 280°

MODEL NO. S-3-3.

NOTE: THE 90° TURN REQUIRES

TWO STROKES PER OPERATION.



RATCHET WHEEL.

90° TURN WITH
100° ± 5° ACTUATOR.

REMOVE (2) TABS

FOR 180° TURN

WITH 280° ACTUATOR.

NAME

K. B. JEE

REFERENCE

A-L Loader

DATE

2-25-80

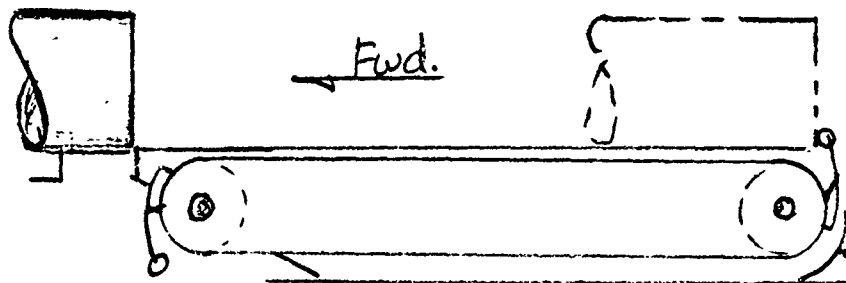
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OF

CARRIAGE LOADER

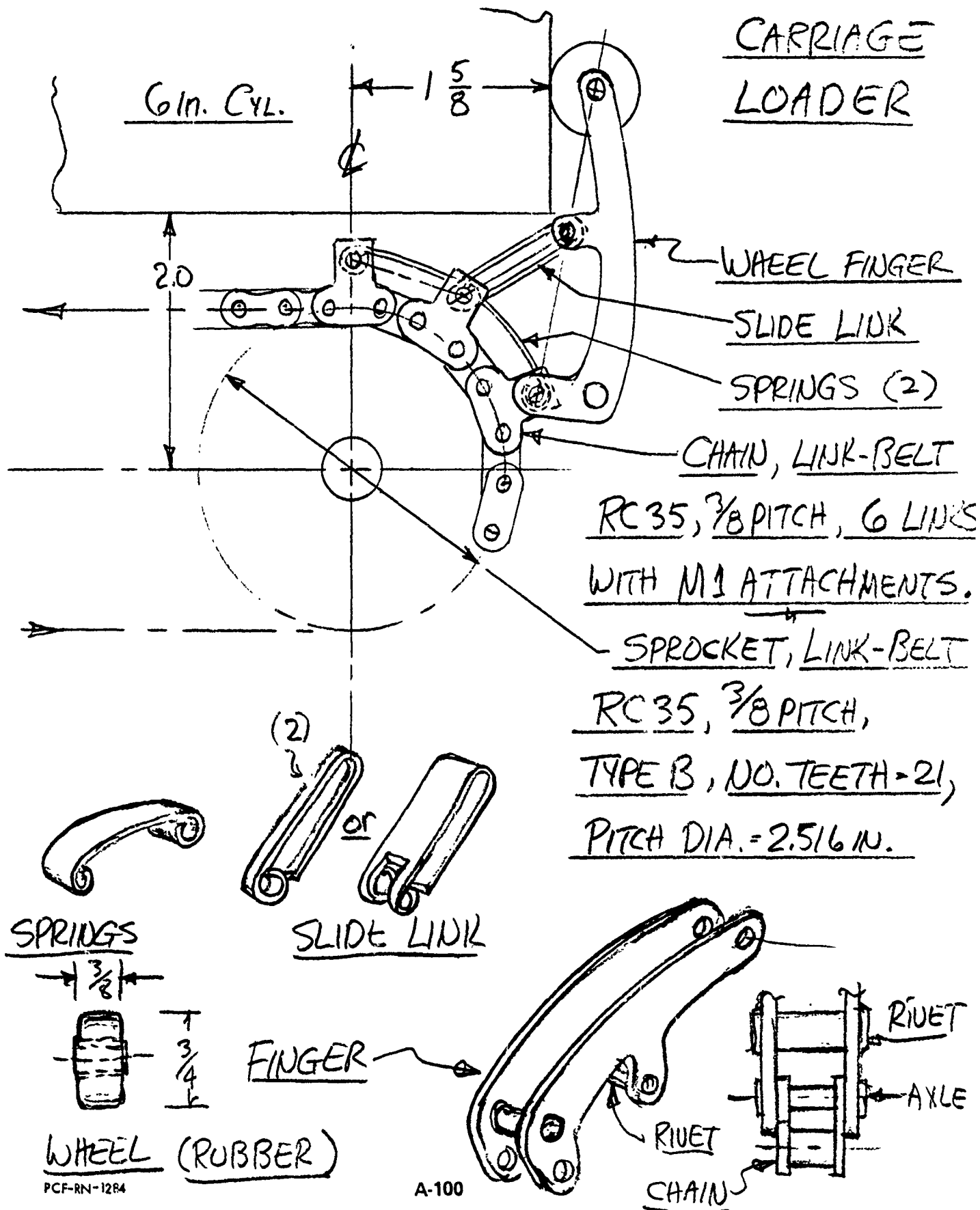
THE SCHEME DRAWN UTILIZES TWO
ROLLER CHAIN LOOPS. THERE ARE TWO
FINGER-LINK ASSEMBLIES ON EACH CHAIN,
EACH HALFWAY AROUND. FOR THIS REASON
THE CHAIN TRAVELS ONLY IN ONE DIRECTION:
WHEN THE FINGER PUSHING THE ROUNDS OUT
COMPLETES ITS JOB, THE OTHER FINGER
WILL BE IN POSITION FOR THE NEXT LOAD.
ALSO, THE ROUNDS MAY BE MOVED FWD. INTO
THE FWD. TRAY AREA HAVING THE COVER OVER
IT. THEN WHEN THE CARRIAGE RETURNS TO
ITS LOADING POSITION, ONLY A SHORT PUSH
WILL BE REQ'D. TO LOAD THE ROUNDS.



CAM SURFACE TO
COLLAPSE FINGER.
(SPACE REQ'D.)

NAME KBouee ENC
DATE 2-25-80

REFERENCE A-L. LOADER
PAGE 100 OF



PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

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REFERENCE


A-L Loader

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OF

CARRIAGE LOADER.

THE THREE (3) LINKS WITH THE M1 ATTACHMENTS HOLDING THE WHEEL FINGER, ETC., ARE CONNECTED BY SPRINGS.  THESE WILL TEND TO STABILIZE THE TRIO & MINIMIZE TIPPING (OR BUCKLING) OF THE LINKS WHEN UNDER LOAD. IT IS NOT NECESSARY TO HAVE A WHEEL ON THE END OF THE FINGER. A PLATE OR BAR, ETC., COULD BE SUBSTITUTED & THE NECESSARY CHANGES MADE IN THE SUPPORT STRUCTURE SHOWN.

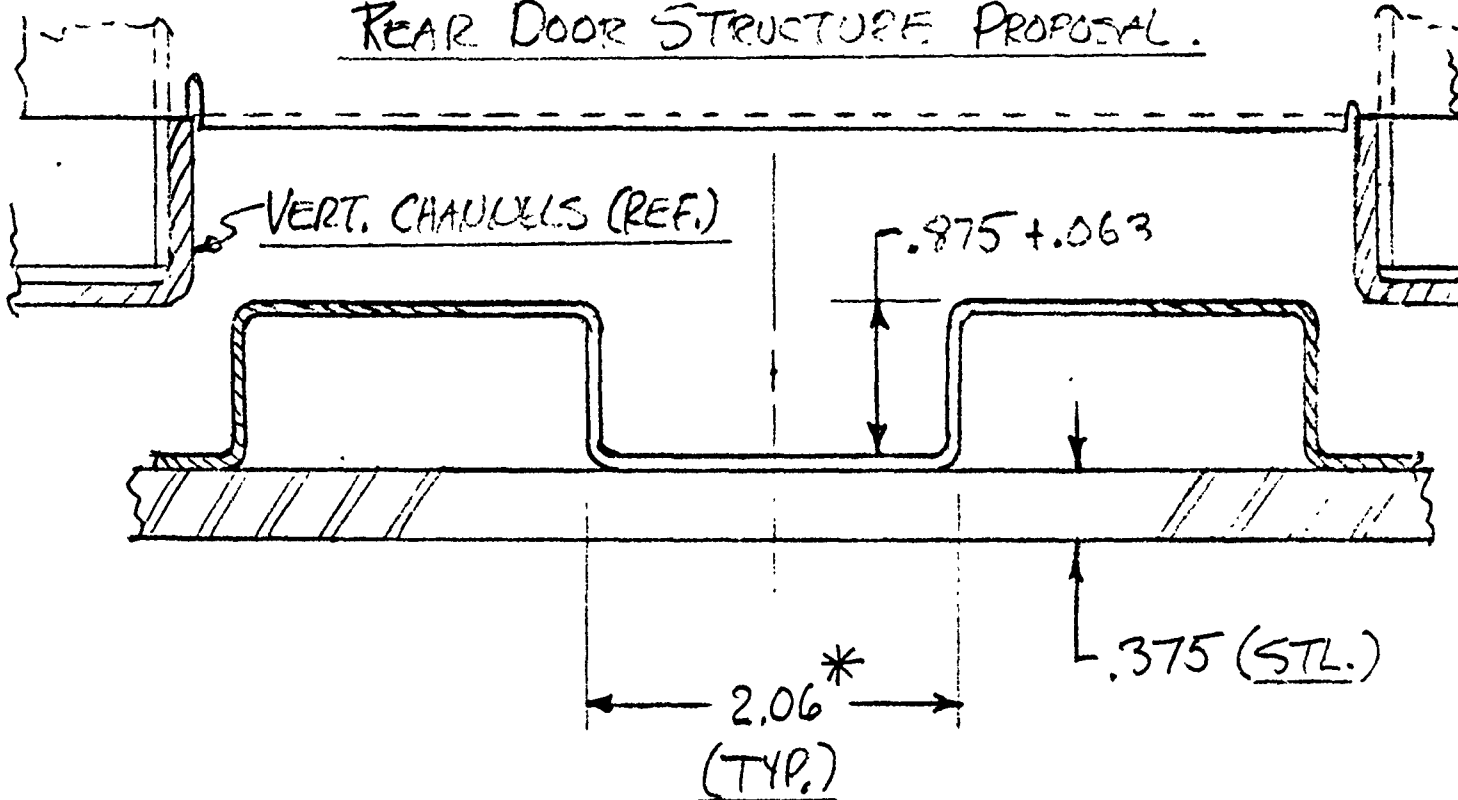
THE WHEEL FINGER-SLIDE LINK ARRANGEMENT WILL NOT ONLY COLLAPSE TO MEET THE REQMTS. AT THE REAR OF THE BOX, BUT WILL PERMIT THE PUSHING OF THE ROUNDS PAST THE END OF THE LOADING TRAY ONTO THE CARRIAGE WITHOUT THE CHAIN OR SPROCKET INTERFERING WITH THE CARRIAGE STRUCTURE.

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME K. BOE
DATE 2-27-80

REFERENCE A-L. MAGAZINE
PAGE 102 OF

REAR DOOR STRUCTURE PROPOSAL.



THE DOOR COULD BE MADE FROM .375 T-1 STL.
OR EQUIV. (FOR ARMOR PROTECTION REQTS.) & THE
CORRUGATION OF 18 GA. (.0478) STEEL SHEET.
THE BOX OPENING = 8.055 - 1.5 = 6.555 IN. ON THE
AVERAGE. THEREFORE - $\frac{8.055}{4} = 2.01375$ " C-C DIST.
(CORRUGATIONS)

* $2.01375 + .0478 = 2.06155$ ". THIS SPACING WOULD
ALLOW FOR SOME DIMENSIONAL ERROR.

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME K. BOEER
DATE 2-29-80

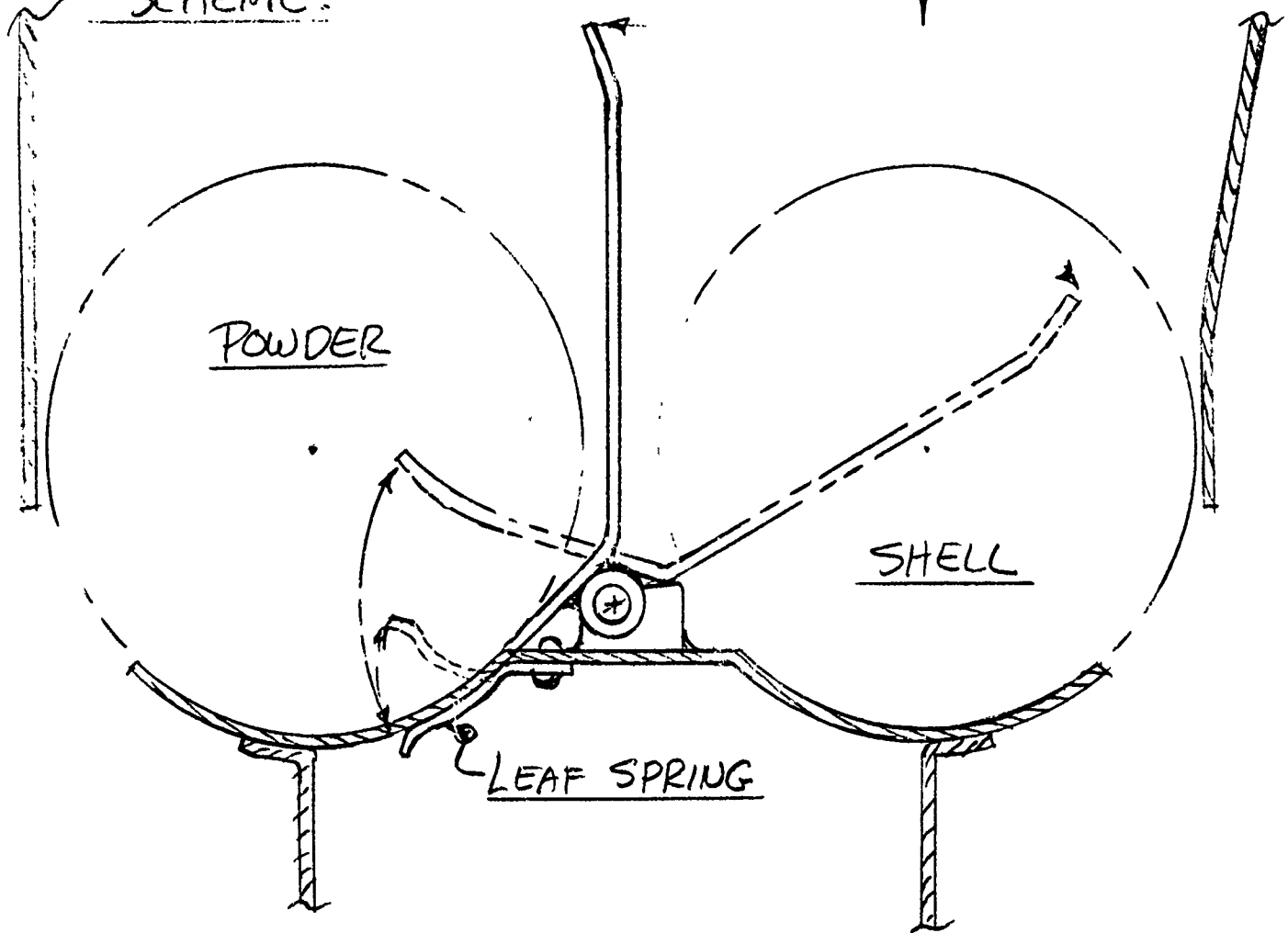
REFERENCE A-L LOADER
PAGE 103 OF

AUTOMATIC LOAD SWITCH

THE POWDER CANNISTER
IS DROPPED FIRST IN THIS

SCHEME.

DESCENDING
LOAD
PATH



SECTIONAL VIEW
LOOKING FWD.

NOTE: THE HINGE SHOULD
BE LOOSE ENOUGH SO THAT
STICKING OR BINDING IS IM-
POSSIBLE.

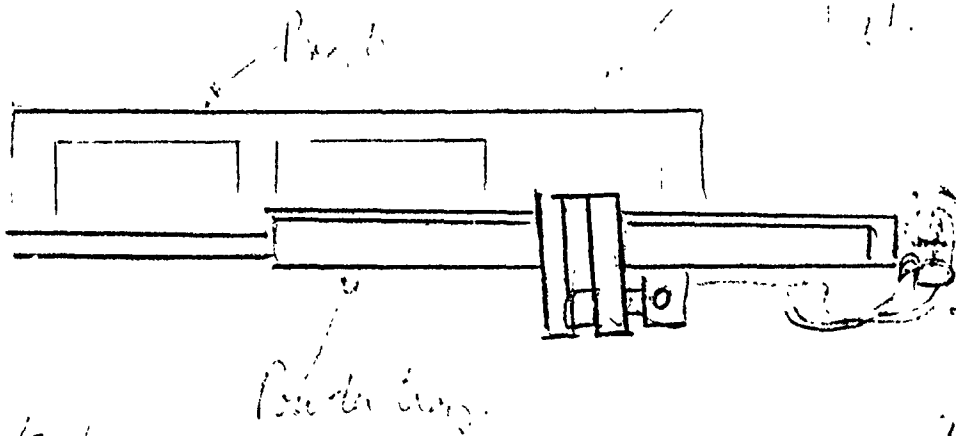
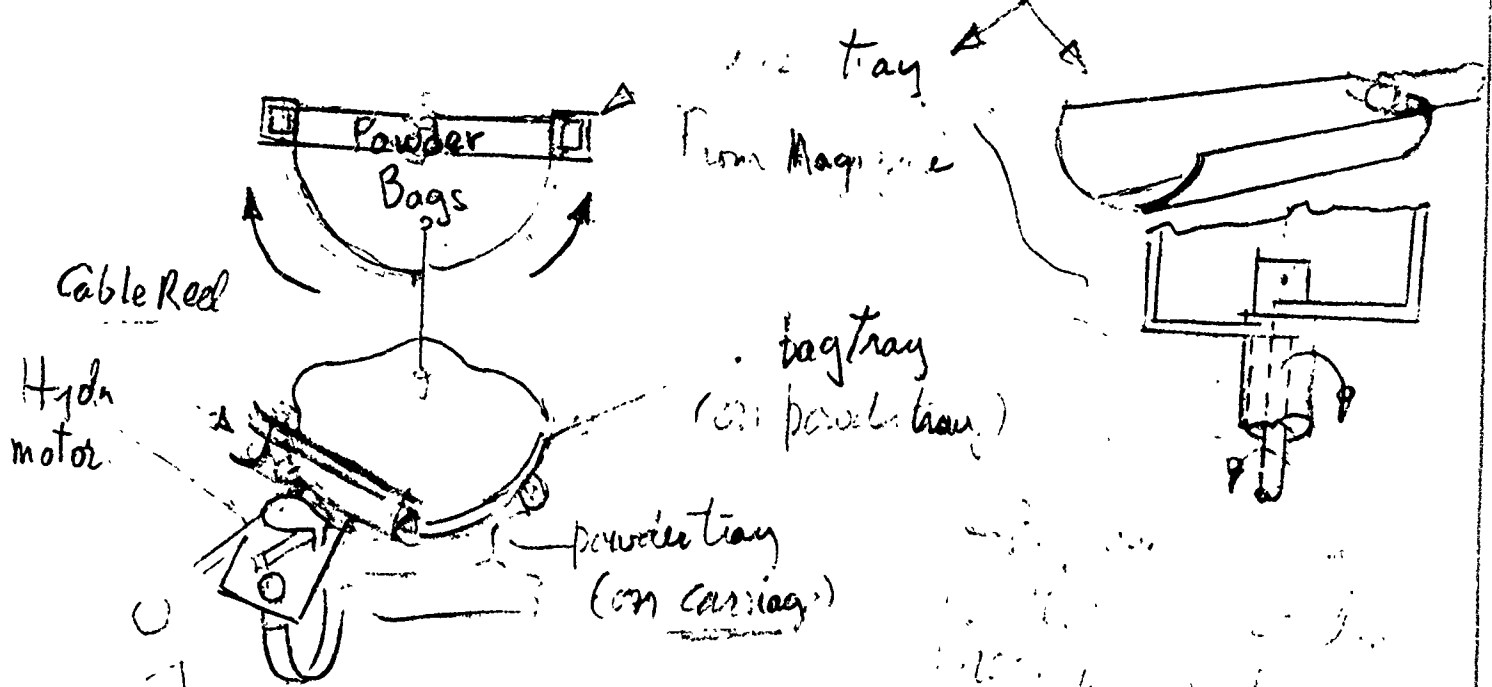
To load bags of powder
without combustible case

(148)

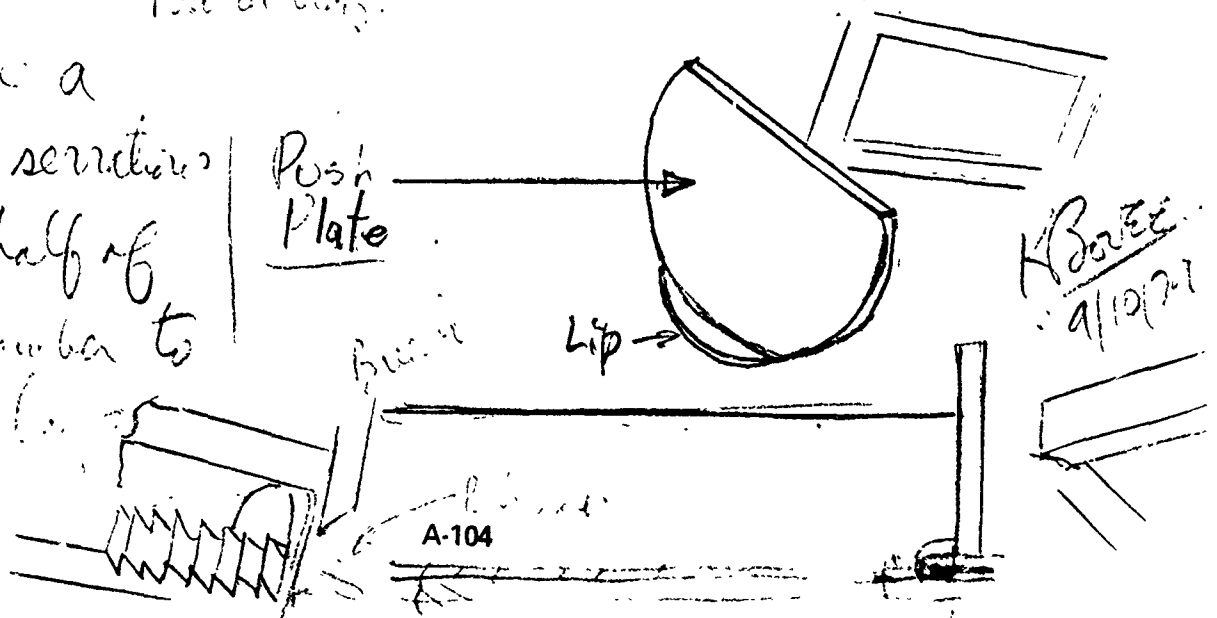
9/10/79

(104) Idea System

Use dumper tray



Has to be a
series of serrations
in lower half of
powder chamber to
prevent bagging



A-104

K. B. B. E.
9/10/79

APPENDIX

- B1 M109 Recoil Cylinders**
- B2 Orifice Area Derivation**
- B3 Vehicle Motion Resulting from Firing
 Large Weapons**
- B4 Stress Calculations**

APPENDIX B1

M109 RECOIL CYLINDERS

Tentative parameters (Telecon Walter Pape November 21, 1979)

Projectile velocity 3250 ft/sec

Projectile weight 98 lb

Powder weight 40 lb

Recoiling weight 9600 lb

$$\text{Impulse} = \frac{(1.3) (3250) (98 + \frac{40}{2})}{1} = 15483 \text{ lb /sec}$$

$$\text{Free recoil energy} = \frac{\frac{I^2}{2m}}{(2) (9600)} = \frac{(15483)^2 (32.2)}{(2) (9600)} = 402,036 \text{ ft-lb}$$

Nominal reaction for 21-inch recoil

$$R = \frac{(0.8) (402036) (12)}{21} = 184,000 \text{ lb}$$

$$\text{Nominal time for recoil} = \frac{15483}{184000} = 0.084 \text{ sec}$$

Use 200,000 lb reaction for design.

Maximum O.D. of cylinder = 6.5 in. (For reasonable retrofit)

Assume piston rod diameter = 2.5 in.

Assume 3/8 cylinder wall and 1/4 sleeve

Trial piston diameter = 6.5 - 1.25 = 5.25 in.

$$\text{Piston area} = \frac{\pi}{4} (5.25^2 - 2.5^2) = 16.739 \text{ sq in.}$$

$$\text{Nominal pressure} = \frac{100000}{16.739} = 5974 \text{ psi}$$

$$\text{Hoop stress} = \frac{(5974) (5.75)}{.75} = 45,800 \text{ psi (OK)}$$

$$\text{Orifice area} = \frac{A_p}{K} \sqrt{\frac{N A_p e S}{WC}} \text{ (See Appendix B2)}$$

Where:

- A_p = Recoil piston area sq in.
- N = Number of recoil cylinders
- e = Recoil oil density lb/cu in.
- S = Distance to end of recoil in.
- W = Recoiling weight lb
- K = Orifice discharge coefficient
- C = Ratio of orifice generated resistance to total resistance to recoil.

Use a sharp edge orifice because it is influenced less by variations in viscosity than a round edge and can live better with contaminants. The discharge coefficient "K" for a sharp edge orifice is .61.

For guns whose trunnion reaction are very high compared to their weight, item "C" is very nearly equal to 1.0 and can be ignored in the initial design.

Then the approximate maximum orifice area =

$$A_o = \frac{16.739}{.61} \sqrt{\frac{(2) (16.739) (.0313) (21)}{9600}}$$

$$A_o = 1.314 \text{ sq in.}$$

$$\text{Orifice sleeve area} = \frac{\pi}{4} (5.75^2 - 5.25^2) = 4.320 \text{ sq in.}$$

$$\text{Percent cutout} = \frac{1.314}{4.320} = 30\% \quad (\text{OK})$$

The classic expression for orifice area derived in Appendix B2 will give precise values for the portion of recoil stroke coming after the chamber pressure has ceased to produce a significant force on the breech. For weapons with a relatively long recoil stroke, this expression is all that is needed since the travel consumed while the recoil pressure is building up to maximum is a small part of the total travel.

For weapons with a relatively short recoil stroke, like this one, the recoil travel consumed during the time the weapon is being accelerated can be a significant part of total travel. To optimize the recoil system (i.e. minimum trunnion reaction) for these weapons, the orifice calculations should take into account the varying breech force.

A very successful solution to this is to determine the position in recoil, where the net force on the gun is zero and apply the classic orifice area formula from there to end of recoil. Then hold the orifice area, at the zero force point, constant to the beginning of recoil.

Another method is to actually solve for the recoil velocity at small recoil increments during the time of varying breech force. This approach has been made much easier with the advent of the computer. Following are calculations for orifice area using this method.

The interior ballistics were not available for the weapon with the parameters used for this design. A computer printout of the interior ballistics of a similar weapon was available and is used here for an interim orifice design until the interior ballistics are finalized.

1. 10600-RTED. HYPERID DIV.

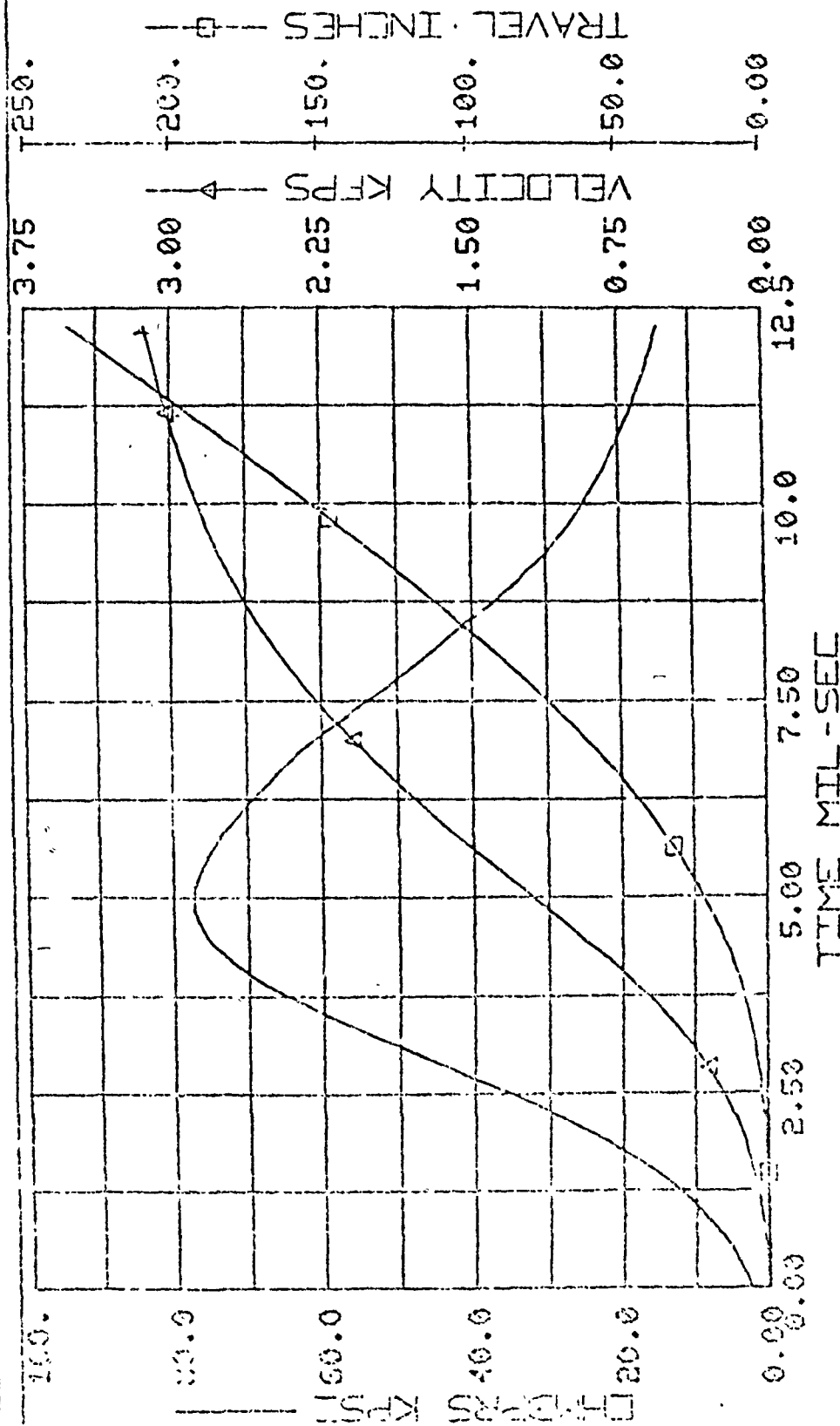
2. 106 SPH 018P

3. 106 NO. 5

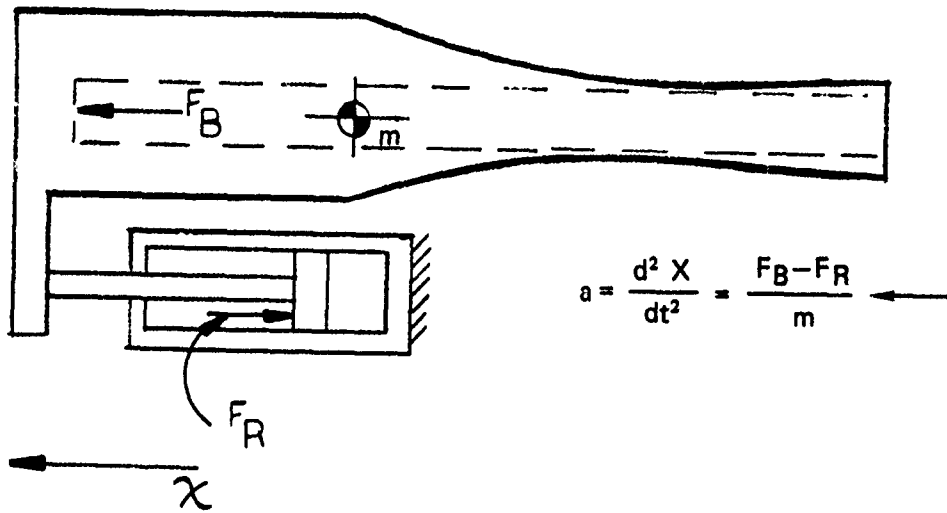
4. 106 0.305

5. 106 1.165

IBIS UPDATED: 9-22-78
HRA: N/A DATE: 11-22-78
GUN: 155MM SPH
UMUZ: 3.127 KFPS
PCMAX: 77.27 KPSI



RECOIL ORIFICE



Let:

- I = Solution point
- t = Time --- sec
- m = Recoiling mass --- $\frac{\text{lb}}{\text{sec}^2}$
in.
- X = Recoil travel --- in.
- V_g = Recoil velocity --- in/sec
- F_B = Breech force --- lb
- F_R = Recoil force --- lb
- F_m = Mass force = F_B - F_R
- A_p = Recoil piston area --- sq in.
- A_o = Orifice area --- sq in.
- K = Orifice discharge coefficient
- b = Orifice width --- in.

$$V_{oil} = \sqrt{2g \frac{P}{e}} = V_g \frac{A_p}{KA_o}$$

$$A_o = \frac{V_g A_p}{K \sqrt{2g \frac{P}{e}}}$$

$$P = \frac{F_R}{2 A_p} \quad (2 \text{ cylinders})$$

$$A_o = \frac{V_g A_p}{K \sqrt{2g \frac{F_R}{2 A_p e}}} = \frac{V_g A_p}{K} \frac{\sqrt{A_p e}}{\sqrt{g F_R}}$$

$$A_o = \frac{1}{K} \sqrt{\frac{A_p^3 e}{g}} \frac{V_g}{\sqrt{F_R}} \quad \leftarrow$$

$$A_p = 16.739 \text{ sq in.}$$

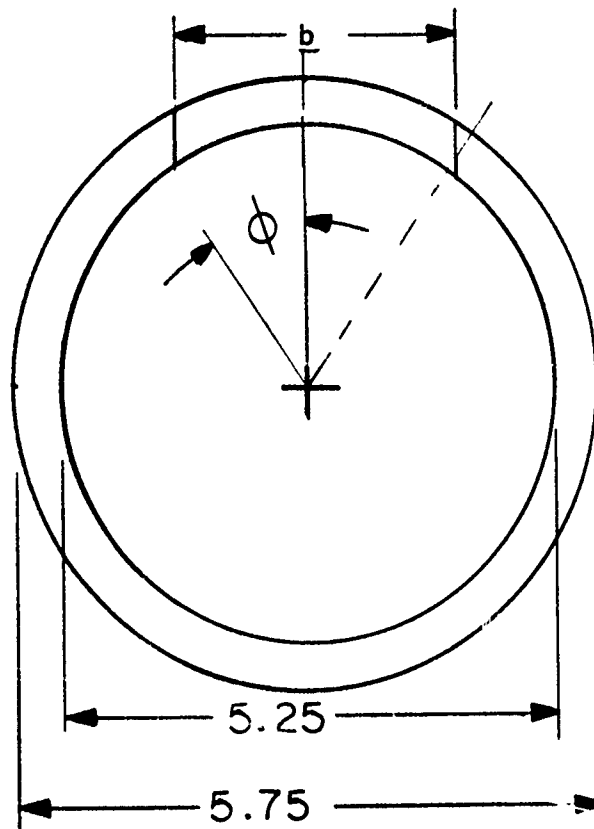
$$e = .0313 \text{ lb/cu in.}$$

$$g = 386 \text{ in/sec}^2$$

$$K = .61$$

$$A_o = \frac{1}{.61} \sqrt{\frac{(16.739)^3 (.0313)}{386}} \frac{V_g}{\sqrt{F_R}}$$

$$A_o = 1.011 \frac{V_g}{\sqrt{F_R}}$$



$$\text{Sleeve section area} = \frac{\pi}{4} (5.75^2 - 5.25^2) = 4.320 \text{ sq in.}$$

Using 3 slots per sleeve

$$\phi = \frac{2 \pi A_o}{(6) (4.320)} = .2424 A_o$$

$$\phi = (.2424) (1.011) \frac{V_g}{\sqrt{F_R}} = .245 \frac{V_g}{\sqrt{F_R}}$$

$$b = \frac{(5.75 + 5.25)}{2} \sin \phi = 5.5 \sin \phi$$

$$b = 5.5 \sin .245 \frac{V_g}{\sqrt{F_R}}$$

Solution of basic recoil equation $a = \frac{d^2 X}{dt^2} = \frac{FB - FR}{m}$ and the equation for orifice width $b = 5.5 \sin .245 V_g / \sqrt{FR}$

I	t	X	V _g	a	F _m	FB	FR	A _o	b
1	0.0	0.0	0.0	2360.9	58731.0	58731.0	0.0	0.9000	1.1900
13	0.0057	0.5366	289.7	84596.6	2104427.0	2210310.0	105882.2	0.9000	1.1900
25	0.0110	2.8278	515.0	10086.0	250898.5	580398.5	329500.0	0.9071	1.1992
37	0.0152	5.0827	530.4	-127.8	-3178.6	326321.4	329500.0	0.9341	1.2344
49	0.0186	6.6444	525.1	-2978.2	-74086.2	255413.8	329500.0	0.9249	1.2224
61	0.0224	8.8149	507.2	-6380.7	-158725.2	170774.8	329500.0	0.8933	1.1814
73	0.0238	9.5326	497.3	-7375.1	-183464.3	146035.8	329500.0	0.8760	1.1588
85	0.0261	10.6377	479.5	-8249.8	-205223.3	124276.7	329500.0	0.8445	1.1178
97	0.0266	10.8934	475.1	-8348.0	-207664.7	121835.3	329500.0	0.8367	1.1076
109	0.0269	11.0344	472.6	-8398.0	-208909.8	120590.2	329500.0	0.8323	1.1018
121	0.0272	11.1886	469.8	-8447.3	-210135.1	119364.9	329500.0	0.8274	1.0955
133	0.0274	11.2863	468.0	-8479.4	-210933.1	118566.9	329500.0	0.8243	1.0914
145	0.0277	11.4252	465.5	-8533.4	-212275.9	117224.1	329500.0	0.8199	1.0856
157	0.0280	11.5358	463.5	-8582.8	-213505.8	115994.3	329500.0	0.8163	1.0809
169	0.0282	11.6154	461.9	-8625.9	-214578.4	114921.6	329500.0	0.8136	1.0773
181	0.0283	11.7007	460.4	-8678.4	-215883.1	113616.9	329500.0	0.8108	1.0738
193	0.0285	11.7759	458.9	-8731.2	-217198.4	112301.6	329500.0	0.8083	1.0705
205	0.0286	11.8168	458.2	-8767.4	-218098.7	111401.3	329500.0	0.8069	1.0687
217	0.0287	11.8917	456.7	-8833.5	-219742.1	109757.9	329500.0	0.8044	1.0653
229	0.0289	11.9460	455.7	-8889.1	-221126.0	108374.0	329500.0	0.8025	1.0629
241	0.0290	12.0137	454.3	-8964.4	-222998.3	106501.8	329500.0	0.8002	1.0598
253	0.0292	12.0679	452.8	-9064.9	-225497.6	104002.4	329500.0	0.7976	1.0564
265	0.0293	12.1283	452.0	-9122.2	-226924.4	102575.6	329500.0	0.7961	1.0546
277	0.0294	12.1686	451.2	-9185.1	-228489.9	101010.1	329500.0	0.7947	1.0527
289	0.0294	12.2088	450.4	-9250.4	-230113.4	99386.6	329500.0	0.7932	1.0508
301	0.0295	12.2490	449.6	-9324.6	-231959.8	97540.3	329500.0	0.7918	1.0488
313	0.0296	12.2957	448.6	-9412.8	-234153.9	95346.1	329500.0	0.7901	1.0466
325	0.0297	12.3424	447.6	-9515.0	-236696.1	92803.9	329500.0	0.7883	1.0443
337	0.0298	12.3889	446.6	-9620.2	-239311.4	90188.6	329500.0	0.7866	1.0420
349	0.0300	12.4426	445.4	-9752.4	-242599.7	86900.3	329500.0	0.7845	1.0393
361	0.0333	13.8869	411.5	-10486.6	-260864.7	68635.3	329500.0	0.7247	0.9610
373	0.0391	16.0667	348.4	-11499.8	-286068.1	43431.9	329500.0	0.6136	0.8149
385	0.0448	17.8597	280.8	-12119.4	-301481.1	28018.9	329500.0	0.4945	0.6575
397	0.0505	19.2637	210.4	-12496.6	-310865.6	18634.3	329500.0	0.3705	0.4931
409	0.0562	20.2601	138.2	-12731.9	-316718.6	12781.4	329500.0	0.2435	0.3243
421	0.0619	20.8409	65.0	-12887.3	-320584.0	8916.0	329500.0	0.1145	0.1526

STOP

PROCESSING

48 UNITS

BUFFER

Piston rod area required to pull 100,000 pounds at 50,000 psi stress

Area = 2 sq in.

Rod diameter is 2.5 in.

Maximum inside diameter of rod = d

$$2 = \frac{\pi}{4} (2.5^2 - d^2)$$

$$d = 1.924 \text{ use } 1\text{-}3/4$$

$$\text{Buffer spear area} = \frac{\pi}{4} (1.75^2) = 2.405 \text{ sq in.}$$

Assume the counter recoil force in battery will be 1-1/2 times the recoiling weight (maximum) and the pressure will double in the 21" stroke.

$$\text{Approximate stored energy} = 9600 \left[1.5 + \left(\frac{3}{2} \right) \right] 21$$

$$E_n = 453,600 \text{ in-lb}$$

The recoil cylinders will dissipate approximately 1/2 of this during counterrecoil.

Energy to be dissipated by 2 buffers is then 226,800 in-lb

Force per buffer with a "6" stroke

$$F = \frac{201000}{(2)(6)} = 18,900 \text{ lb}$$

$$\text{Pressure} = \frac{16750}{2.405} = 7,859 \text{ psi}$$

$$\text{Buffer orifice area} = \frac{A_p}{K} \sqrt{\frac{A_p N e X}{W}} \quad (\text{See Appendix B2})$$

Use a round edge orifice, $K = 1$, because the small clearance between the I.D. of the buffer cavity and the buffer spear precludes a sharp edge orifice.

$$\text{Then } A_o = \frac{2.405}{1} \sqrt{\frac{(2.405)(2)(.0313) X}{9600}}$$

$$A_o = .00952 \sqrt{X}$$

Use 3 orifice grooves .100 wide

$$\text{Groove depth} = \frac{.00952 \sqrt{X}}{(3)(.100)} = .03173 \sqrt{X}$$

$$\text{Buffer orifice depth } d = .03173 \sqrt{X}$$

X	d
in	in
0	0
1	.032
2	.045
3	.055
4	.063
5	.071
6	.078

ORIFICE AREA DERIVATION

Find the orifice area for a constant force recoil system.

Let:

- W = Recoiling weight — — — lb
 A_p = Recoil piston area — — — sq in.
 N = Number of recoil cylinders
 e = Recoil oil density — — — lb/cu in.
 S = Distance to end of recoil — — — inches
 F = Total force resisting recoil — — — lb
 P = Pressure developed by orifice — — — psi
 C = Ratio of orifice generated force to total force
 K = Orifice discharge coefficient
 V_g = Recoil velocity — — — in./sec
 V_o = Velocity of oil through orifice — — — in./sec
 A_o = Orifice area — — — sq in.
 G = Acceleration of gravity — in./sec²

After the propellant gasses cease to act on the breech, the kinetic energy of the recoiling weight at any point "S" is equal to the work that will be done by the constant force "F" acting through the distance "S".

$$K.E. = FS = \frac{WV_g^2}{2G}$$

$$V_g = \sqrt{\frac{2GFS}{W}}$$

$$\text{Oil velocity, } V_o = V_g \frac{A_p}{KA_o}$$

V_o also equals

$$\sqrt{2G \frac{P}{e}}$$

$$\therefore \sqrt{2G \frac{P}{e}} = V_g \frac{A_p}{KA_o} = \sqrt{\frac{2GFS}{W}} \frac{A_p}{KA_o}$$

$$A_o = \frac{A_p}{K} \sqrt{\frac{2GFS}{W}} \sqrt{\frac{1}{2G \frac{P}{e}}}$$

$$A_o = \frac{A_p}{K} \sqrt{\frac{FS e}{WP}}$$

$$\text{Since } C = \frac{PNA_p}{F} \quad P = \frac{F}{NA_p C}$$

$$\therefore A_o = \frac{A_p}{K} \sqrt{\frac{NA e S}{WC}} \longleftarrow$$

VEHICLE MOTION RESULTING FROM FIRING LARGE WEAPONS

Chase I suspension is active. This is analogous to a load suddenly applied to a mass spring system where the product of the magnitude of the load, firing reaction, and its duration, time of recoil, is a constant which is equal to the impulse of the round fired.

Let

I_O = Moment of inertia about the point of rotation

K = Torsional spring rate of suspension about point of rotation

I_m = Impulse of round

F = Trunnion reaction

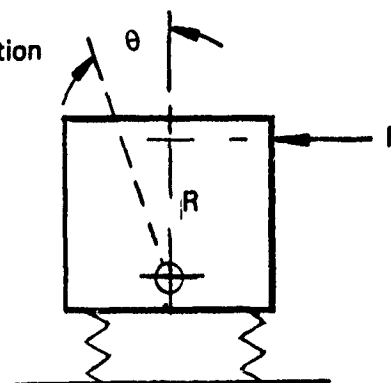
t = Time of recoil

R = Effective lever arm of force F

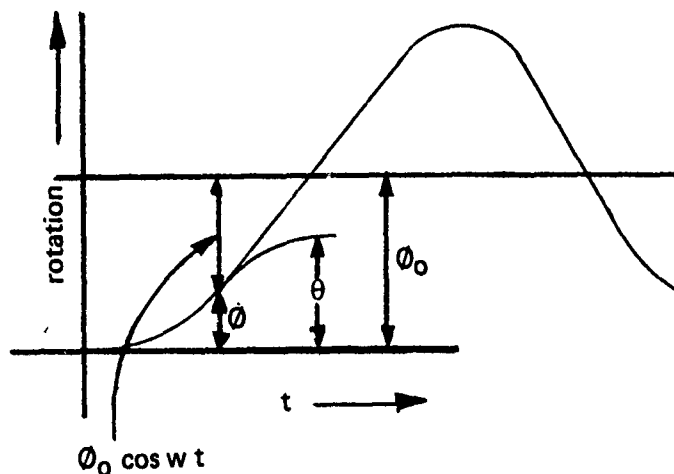
Θ = Total rotation resulting from firing

ϕ_o = Static rotation under load F

ϕ = Rotation when reaction ceases or at time t



Assume No Dampening



$$\phi = \phi_o - \phi_o \cos w t = \phi_o (1 - \cos w t)$$

$$\phi_o = \frac{FR}{K}$$

$$\phi = \frac{FR}{K} (1 - \cos w t)$$

$$\text{Energy input} = FR \phi$$

$$E_n = \frac{F^2 R^2}{K} (1 - \cos w t)$$

At turn around, Θ rotation, all input will be in strain energy of the suspension.

$$\begin{aligned}
 E_n &= 1/2 K \Theta^2 \\
 1/2 K \Theta^2 &= \frac{F^2 R^2}{K} (1 - \cos w t) \\
 \Theta &= \frac{FR}{K} \sqrt{2(1 - \cos w t)} \\
 W &= \sqrt{\frac{K}{I_m I_o}} \\
 t &= \frac{I_m}{F} \\
 \Theta &= \frac{FR}{K} \sqrt{2(1 - \cos \sqrt{\frac{K}{I_o}} \frac{I_m}{F})} \leftarrow
 \end{aligned}$$

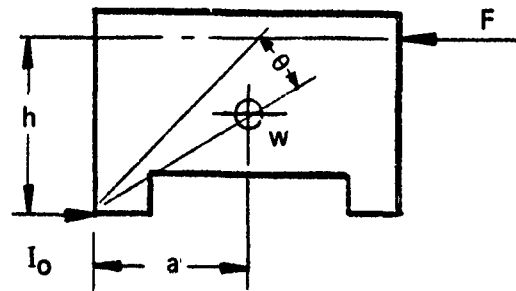
Case II Suspension is locked out

Let

α = Rotational acceleration

w = Vehicle weight

ϕ = Rotation when reaction ceases



Since the angle of rotation will be relatively small, "h" and "a" can be considered as remaining constant without appreciable error.

Then

$$\begin{aligned}
 \alpha &= \frac{Fh - Wa}{I_o} \\
 \phi &= 1/2 \alpha t^2 = \left(\frac{Fh - Wa}{2I_o} \right) t^2 \\
 t &= \frac{I_m}{F} \\
 \phi &= \left(\frac{Fh - Wa}{2I_o} \right) \frac{I_m^2}{F^2}
 \end{aligned}$$

Energy input = $Fh \phi$

At turn around P.E. $\approx Wa \Theta$

Let $Fh \phi = Wa \Theta$

$$\begin{aligned}
 \Theta &= \frac{Fh}{Wa} \phi \\
 \Theta &= \frac{Fh}{Wa} \left(\frac{Fh - Wa}{2I_o} \right) \frac{I_m^2}{F^2} \\
 \Theta &= \frac{I_m^2 h}{2I_o} \left(\frac{Fh - Wa}{FWa} \right) \\
 \Theta &= \frac{I_m^2 h}{2I_o} \left(\frac{h}{Wa} - \frac{1}{F} \right) \leftarrow
 \end{aligned}$$

M109 motion caused by firing impulse as a function of recoil length.

Approximate parameters of the M109

Impulse of round — — — 15,13 lb/sec

Weight — — — — — 53,000 lb

Suspension spring rate

Fore and aft pitch — — — 27×10^6 lb-in /rad

Lateral — — — — — 28×10^6 lb-in /rad

Moment of inertia

Fore and aft

Lateral

About c.g. sprung weight 187×10^6 lb in²

about c.g. sprung wt 36×10^6 lb

About rear corner 1204×10^6 lb in²

about edge of track 365×10^6

Distance

Trunnions to ground 90 in.

c.g. to spade 102 in.

Trunnion to c.g. 36 in.

1/2 width 62 in.

Case I active suspension

Firing forward

$$\Theta = \frac{FR}{K} \sqrt{2(1 - \cos \sqrt{\frac{K}{I_0}} \frac{I_m}{F})}$$

$$I_m = 15483 \text{ lb sec}$$

$$F = \frac{3.864 \times 10^6}{L} \quad L = \text{Recoil length} \quad \text{in.}$$

$$R = 36''$$

$$K = 27 \times 10^6 \text{ lb-in}$$

$$I_0 = 187 \times 10^6 \text{ lb in}^2$$

$$\sqrt{\frac{K}{I_0}} = \sqrt{\frac{27 \times 10^6 (386)}{187 \times 10^6}} = 7.46$$

$$\Theta = \frac{(3.864 \times 10^6) (36)}{27 \times 10^6 L} \sqrt{2(1 - \cos \frac{(7.46)(15483)(L)}{3.864 \times 10^6})}$$

$$\Theta = \frac{5.152}{L} \sqrt{2(1 - \cos .03L)} \quad \text{rad}$$

$$\Theta = \frac{295}{L} \sqrt{2(1 - \cos .03L)} \quad \text{degrees}$$

Case I Active Suspension

Firing over the side

$$\Theta = \frac{FR}{K} \sqrt{2(1 - \cos \sqrt{\frac{K}{I_0}} \frac{I_m}{F})}$$

$$K = 28 \times 10^6 \text{ lb-in./rad}$$

$$I_0 = 36 \times 10^6 \text{ lb in}^2$$

$$\sqrt{\frac{K}{I_0}} = \sqrt{\frac{(28)(386)}{36}} = 17.33$$

$$\Theta = \frac{(3.864 \times 10^6)(36)}{28 \times 10^6 L} \sqrt{2(1 - \cos \frac{(17.33)(15483) L}{3.864 \times 10^6})}$$

$$\Theta = \frac{4.968}{L} \sqrt{2(1 - \cos .0694L)} \text{ rad}$$

$$\Theta = \frac{285}{L} \sqrt{2(1 - \cos .0694L)} \text{ degrees}$$

Recoil Length L	Firing Forward Θ degrees	Firing Over The Side Θ
20	8.72	18.23
25	8.64	17.39
30	8.55	16.40
35	8.45	15.26
40	8.33	14.01

Case II Suspension Locked Out

Firing forward

$$\Theta = \frac{I_m^2 h}{2I_0} \left(\frac{h}{Wa} - \frac{1}{F} \right)$$

$$I_m = 15483 \text{ lb sec}$$

$$w = 53000 \text{ lb}$$

$$h = 90 \text{ in.}$$

$$I_0 = 1204 \times 10^6 \text{ lb in}^2$$

$$W = 53000 \text{ lb}$$

$$a = 102 \text{ in.}$$

$$F = \frac{3.864 \times 10^6}{L} \quad L = \text{recoil length}$$

$$\Theta = \frac{(15483^2)(90)(386)}{(2)(1204 \times 10^6)} \left(\frac{90}{(53000)(102)} - \frac{L}{3.864 \times 10^6} \right)$$

$$\Theta = 3458 (1.66 \times 10^5 - 2.59 \times 10^7 L)$$

$$\Theta = .03458 (1.66 - .0259 L) \text{ rad}$$

$$\Theta = 1.98 (1.66 - .0259 L) \text{ degrees}$$

Case II Firing over the side

$$\Theta = \frac{I_m^2 h}{2I_o} \left(\frac{h}{Wa} - \frac{1}{F} \right)$$

$$I_o = 365 \times 10^6 \text{ lb in}^2$$

$$a = 62 \text{ in.}$$

$$\Theta = \frac{(15483)^2 (90) (386)}{(2) (365 \times 10^6)} \left(\frac{90}{(53000) (62)} - 2.59 \times 10^{-2} L \right)$$

$$\Theta = 11408 (2.74 \times 10^{-3} - 2.59 \times 10^{-7} L)$$

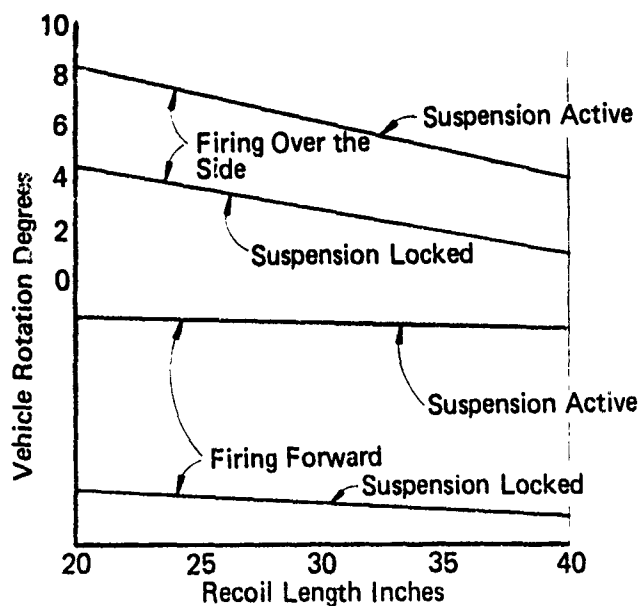
$$\Theta = .11408 (2.74 - .0259 L) \text{ rad}$$

$$\Theta = 6.54 (2.74 - .0259 L) \text{ degrees}$$

Case II Suspension Locked out

Recoil Length L	Firing Forward Θ	Firing Over The Side Θ
20	2.26	14.53
25	2.00	13.68
30	1.75	12.84
35	1.49	11.99
40	1.24	11.14

M109 Rotation from Firing a 15483 Lb/Sec Round



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DATE

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REFERENCE

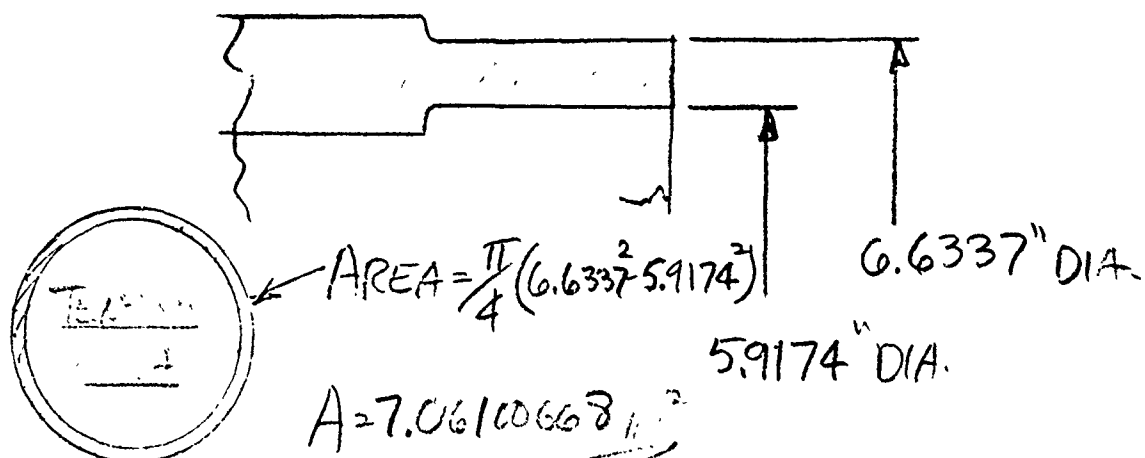
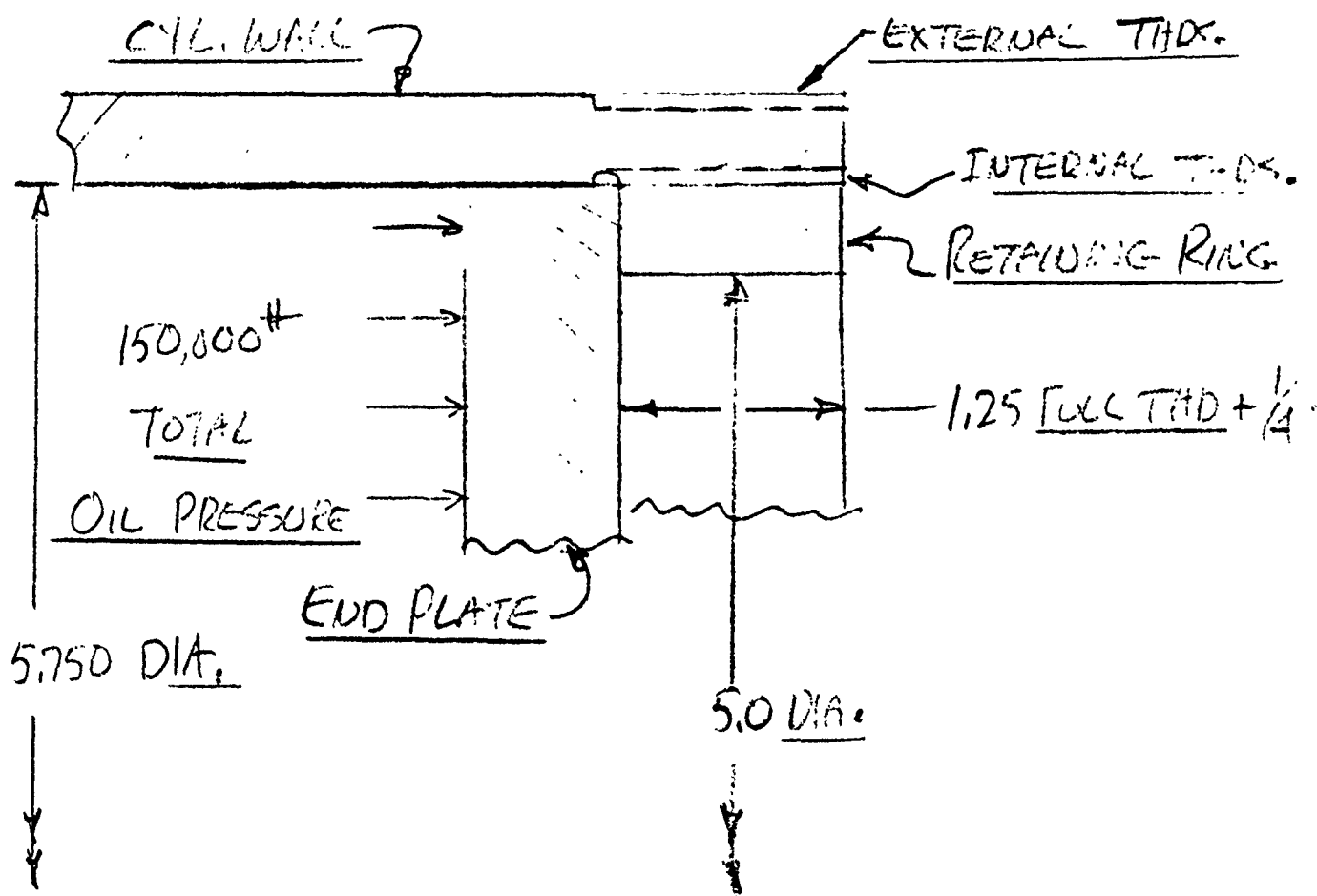
K. B. B. E. Cyl.

PAGE

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OF

STRESS CHECK
ON
THREADS & WALL THICKNESS.



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REFERENCE RECOIL CYC.
PAGE 2 OF

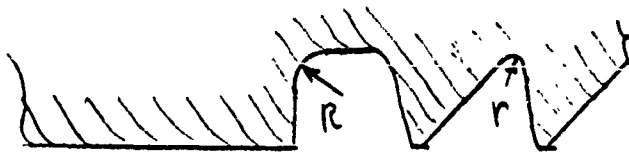
STRESS CHECK.

$$\text{TENSILE STRESS} = \frac{150000}{7.06100668} = 21,243.42984$$

USE 21243.43 PSI

STRESS CONCENTRATION CALCS.

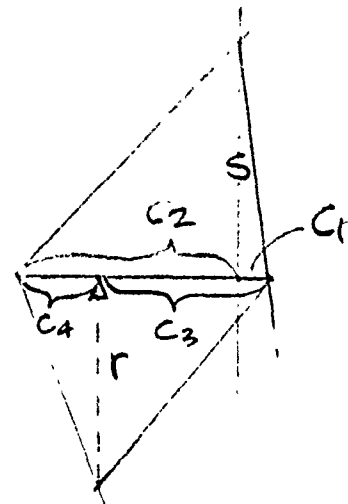
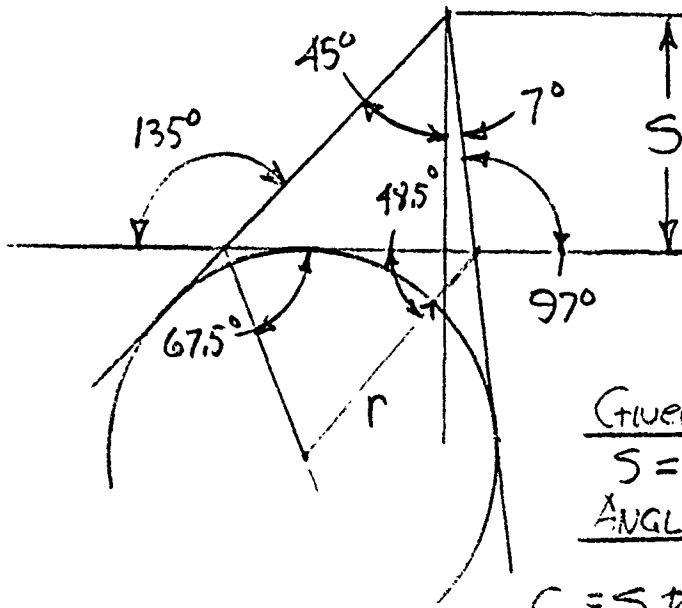
NOTCH EFFECTS



BUTTRESS THREADS

GEOMETRY:

FIND: r



Given:

$$S = .0069$$

ANGLES, 7°, 45°

$$C_1 = S \tan 7^\circ, C_2 = S \tan 45^\circ$$

$$C = C_1 + C_2 = C_3 + C_4$$

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STRESS CHECK.

$$V = C_3 \tan 48.5^\circ = C_4 \tan 67.5^\circ$$

$$C_3 = C_4 \frac{\tan 67.5^\circ}{\tan 48.5^\circ}$$

$$C_4 \left(1 + \frac{\tan 67.5^\circ}{\tan 48.5^\circ} \right) = C = S (\tan 7^\circ + \tan 45^\circ)$$

$$C_4 (1 + 2.135915733) = .0077472134$$

$$C_4 = .0024704788$$

$$C_3 = .0052767345$$

$$V = .0059642634 \sim \text{USE } \underline{\underline{.006}} \text{ AS NOTCH RAD.}$$

GRAPH BELOW IS FROM PETERSON: STRESS CONCENTRATION
DESIGN FACTORS.

In general, it can be said that our knowledge of notch sensitivity is not very satisfactory. However, it is believed that until better information becomes available use of such curves as in Fig. 10 will provide reasonable design information.

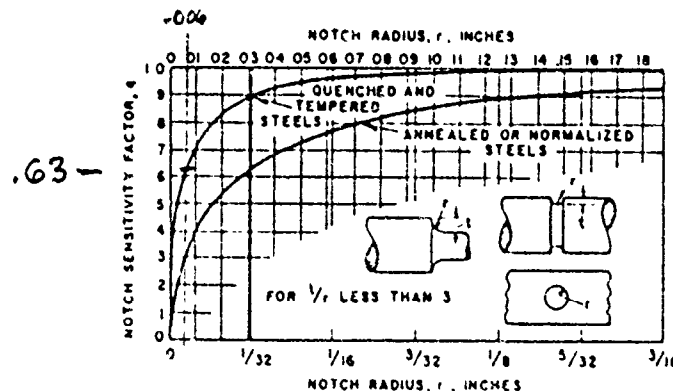


FIG. 10
AVERAGE NOTCH SENSITIVITY CURVES

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STRESS CHECK.

THE FORMULA USED IS:

$$Q = \frac{K_f - 1}{K_t - 1}$$

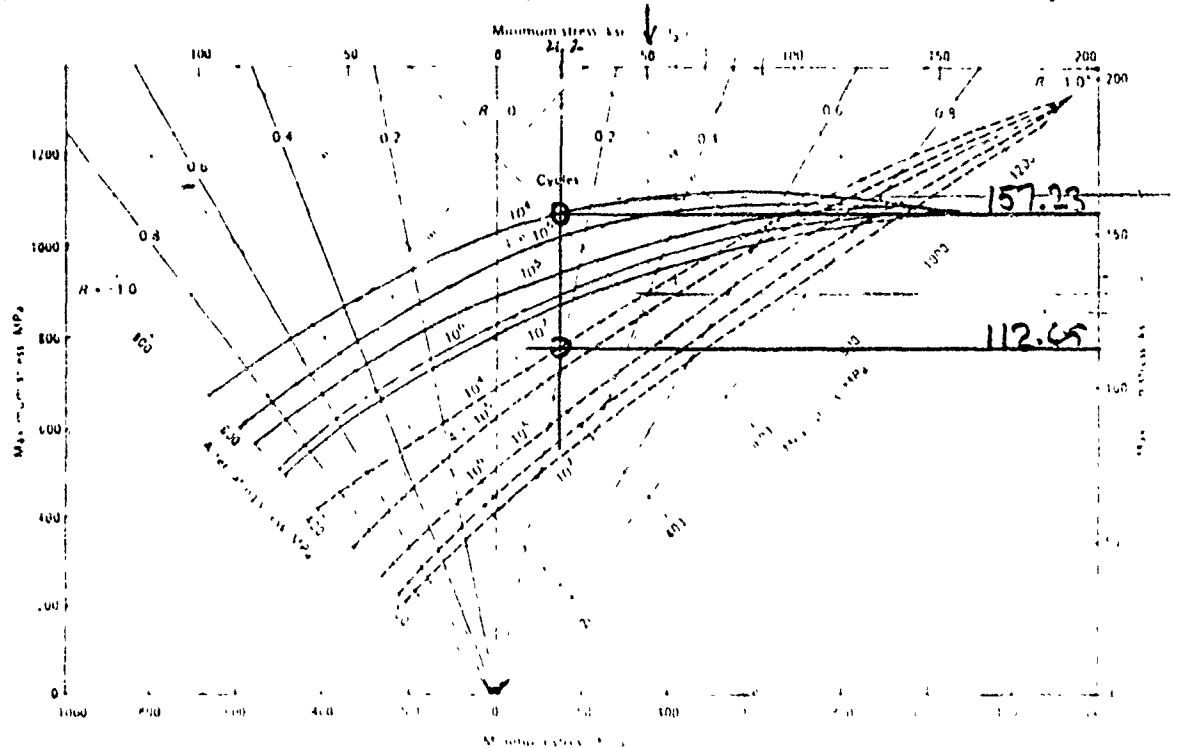
$$K_t = \frac{\sigma_{max.}}{\sigma_{nom.}}$$

$$\sigma_{nom.} = 21243.43 \text{ PSI}$$

(PAGE 2)

$$K_f = \frac{\sigma_e}{\sigma_{nc}} = \frac{\text{FATIGUE STRENGTH OF UNNOTCHED SPEC.}}{\text{FATIGUE STRENGTH OF NOTCHED SPEC.}}$$

Fig. 3 Comparison of constant-lifetime fatigue behavior of notched and unnotched specimens



Constant lifetime fatigue diagram for AISI-SAE 4340 alloy steel (bar), hardened and tempered to a tensile strength of 1035 MPa (150 ksi). Solid lines represent data obtained from unnotched specimens, dashed lines represent data from specimens having notches with $K_t = 3.3$ (Ref 1)

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STRESS CHECK

THE DIAGRAM IS FROM METALS HANDBOOK,
VOL 1, 9th ED. A.S.M. 1978 PG. 667

$$K_f = \frac{157.23}{112.65} = 1.39574.$$

RE-ARRANGING FORMULA:

$$K_t = \frac{(K_f - 1)}{Q} + 1 ; K_t = \frac{.39574}{.63} + 1 = 1.628157$$

$$\sigma_{max} = K_t \sigma_{nom} = 1.628157(21243.43) = 34,587.643$$

PSI.

FOR 4340 STL. -

$$F_{tu} = 150 \text{ KSI.}$$
$$F_{ty} = 135 \text{ KSI.}$$

$$\frac{135000}{34587.643} = \underline{3.9 \text{ FACTOR OF SAFETY.}}$$

THE "R" ON PAGE 2, (SKETCH) SHOULD BE 1/32" MIN.
TO KEEP EVERYTHING NEAT.

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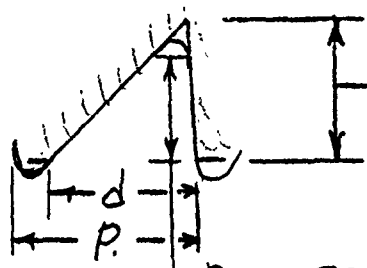
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STRESS CHECK.



$$H_n = .66271 P = .0552$$

$$P = \text{PITCH} = \frac{1}{12} = .08333 \text{ IN.}$$

$$\text{FACE CONTACT } H_e = .0472 \text{ IN.}$$

$$d = P - .00775 = .075583 \text{ APPROX.}$$

$$5.9174 \pi (.075583) = \text{SHEAR AREA FOR (1) THREAD.}$$

$$= 1.4051 \text{ IN}^2$$

$$\text{LOAD} = 90,070 (1.4051) = 126,458.8857 \text{ \# TO}$$

SHEAR OFF ONE THREAD.

$$\text{FACE CONTACT AREA} = \left[(5.85 + .0236)^2 - (5.85 - .0236)^2 \right] \frac{\pi}{4}$$

$$= .4337283 \text{ IN}^2$$

$$\frac{150,070 \text{ \#}}{.4337283} = 345,838.6421 \text{ PSI.}$$

IT IS VITAL THAT MORE THAN ONE (1) THREAD
IS CARRYING THE LOAD. THERE IS A DESIGN
THAT WILL HELP SPREAD THE LOAD ON THE
THREADS.

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OF

STRESS CHECK.

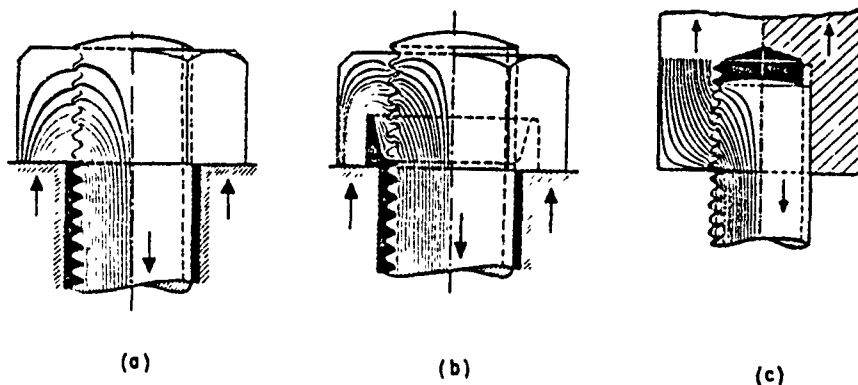
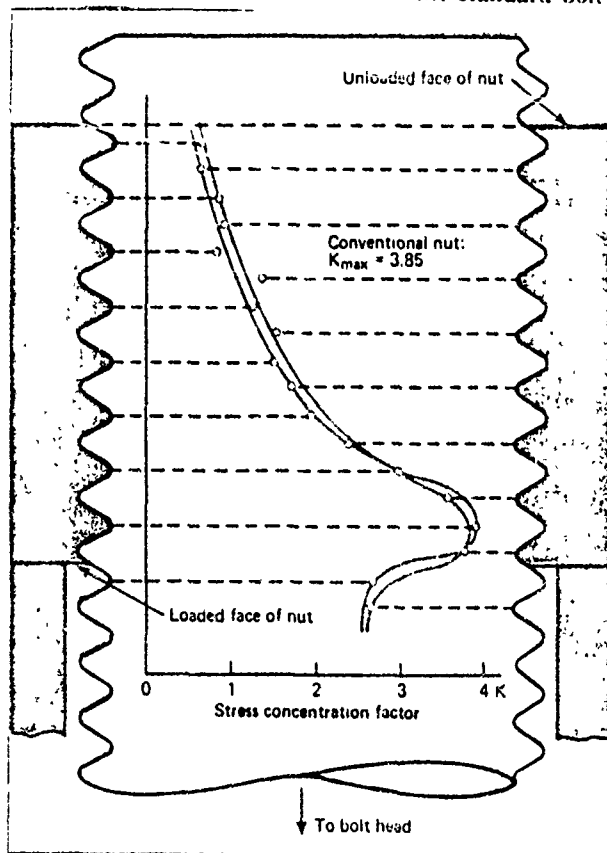


FIG. 97

NUT DESIGNS FATIGUE TESTED (WIEGAND)
(FLOW LINES-HELE SHAW METHOD)

In the arrangement shown in Fig. 97c the transmitted load is not reversed. Fatigue tests¹⁰¹ showed a fatigue strength more than double that of the standard bolt-and-nut combination (Fig. 97a).



The highest loaded threads are those closest to bolt head in normal design. Next drawing shows how load can be evened

ILLUSTRATION FROM
PETERSON-STRESS CONC. ETC.

THIS ILLUSTRATION SHOWS
STRESS FACTOR DISTRIB-
UTION IN BOLT-NUT COMB.

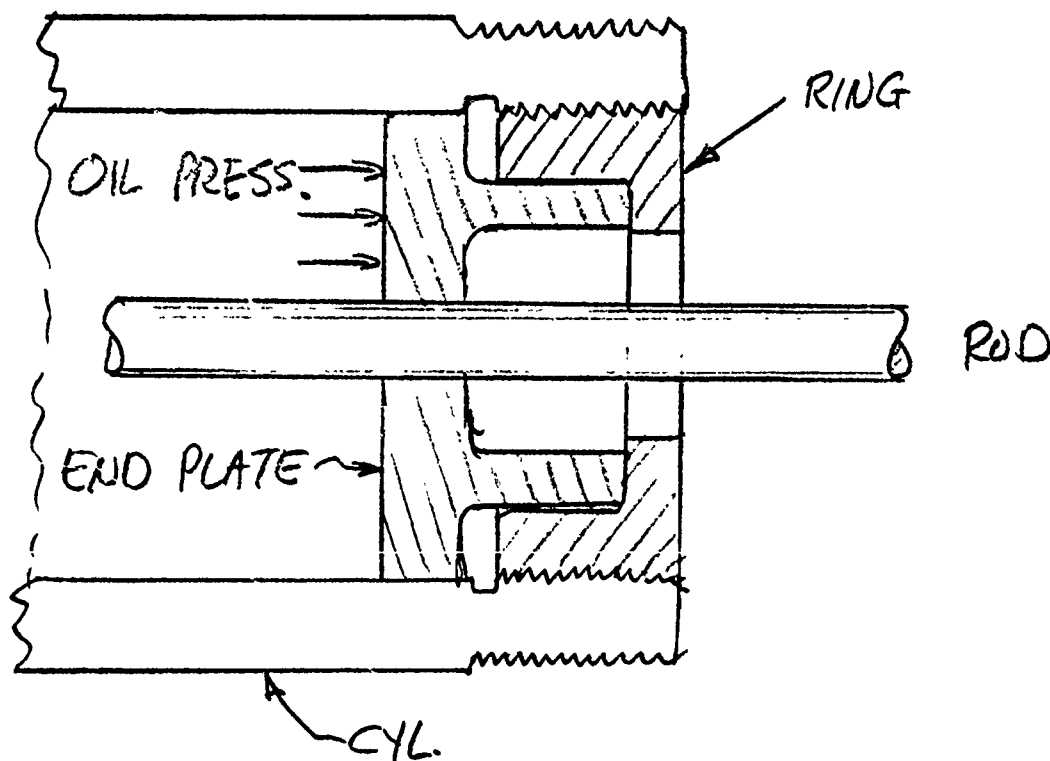
THE ARRANGEMENT
SHOWN IN FIG. 97 (C)
CAN BE ADAPTED FOR
THE PRESENT DESIGN.

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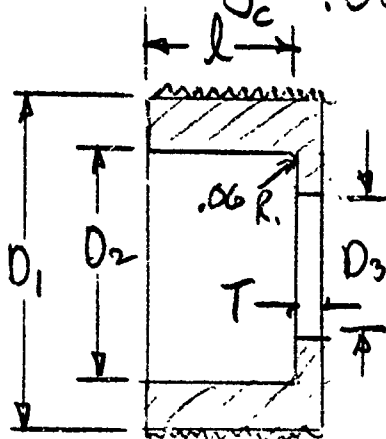
STRESS CHECK.



MATCH RING & CYL. END FOR STRAIN DEFLECTIONS.

$$\delta = e l ; \quad \delta_c = \frac{P l}{A E} = \frac{150000(1.25)}{29500000(7.01600668)} \leftarrow (\text{PAGE 1})$$

$$\delta_c = .00090592 \text{ IN.}$$



$$A_R = \frac{\pi}{4}(D_1^2 - D_3^2) = 7.01600668$$

$$\text{BEARING AREA} = \frac{\pi}{4}(D_2^2 - D_3^2) = \frac{150000(2)}{190,000} = 1.58 \text{ IN}^2$$

$$\text{SHEAR AREA} = \pi D_2 T = \frac{150000(2)}{90000} = 3.33 \text{ IN}^2$$

INTERNAL 5.900 -12 (← UBOIT-2
P

$$G = .0056$$

BASIC MAJOR $\phi = D = \underline{5.9000}$ $\swarrow h$

MIN PITCH $\phi = D - h = 5.9000 - .0500 = \underline{5.8500}$

MAX PITCH $\phi = D - h + Tol = 5.9000 - .0500 + .0084 = \underline{5.8584}$

MIN MINOR $\phi = D - 2h = 5.9000 - (2) .0500 = \underline{5.8000}$

MAX MINOR $\phi = D - 2h + Tol = 5.9000 - (2) .0500 + .0070 = \underline{5.8070}$

MIN MAJOR $\phi = D - 2h + 2h_u = 5.9000 - (2) .0500 + (2) .0552 = \underline{5.9174}$

$$h_e = h - G/2$$

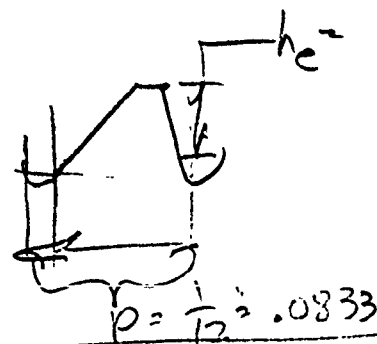
$$.08261 P = 12$$

$$S = .0069$$

$$h_u = .66271 P$$

$$= .0552$$

$$\begin{array}{r} 6.6337 \\ 5.9174 \\ \hline 2 \overline{) 0.7163} \\ .35815 \end{array}$$



EXTERNAL 6.7500 -12 (← UBOIT-2

$\frac{h}{Tol}$

MAX MAJOR $\phi = D - G = 6.7500 - .0059 = \underline{6.7441}$

MIN MAJOR $\phi = D - G - Tol = 6.7500 - .0059 - .0080 = \underline{6.7361}$

MAX PITCH $\phi = D - h - G = 6.7500 - .0050 - .0059 = \underline{6.7391}$

MIN PITCH $\phi = D - h - G - Tol = 6.7500 - .0050 - .0059 - .0089 = \underline{6.7302}$

MAX MINOR $\phi = D - G - 2h_u = 6.7500 - .0059 - (2) .0552 = \underline{6.6337}$

TABLE XIV.4.—Tolerances on Buttress threads, class 1 (free)

Major diameter	Preferred diameters	Threads per inch														Tol on major dia of ext thread and minor dia of int thread
		20	16	12	10	8	6	5	4	3	2½	2	1½	1¼	1	
		Tolerance on pitch diameter, external and internal threads														
in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.
½ to 1½	½, ¾, 1, 1½	0.0063	0.0050	0.0067	0.0111	0.0124	0.0138	0.0154	0.0168	0.0201	0.0224	0.0248	0.0283	0.0303	0.0341	0.0050
1½ to 1	¾, 1, 1½	0.0063	0.0050	0.0067	0.0111	0.0124	0.0138	0.0154	0.0168	0.0201	0.0224	0.0248	0.0283	0.0303	0.0341	0.0050
1 to 1½	¾, 1, 1½	0.0063	0.0050	0.0067	0.0111	0.0124	0.0138	0.0154	0.0168	0.0201	0.0224	0.0248	0.0283	0.0303	0.0341	0.0050
1 to 2½	1½, 2, 2½	0.0063	0.0050	0.0067	0.0111	0.0124	0.0138	0.0154	0.0168	0.0201	0.0224	0.0248	0.0283	0.0303	0.0341	0.0050
2½ to 4	2½, 3, 3½, 4	0.0063	0.0050	0.0067	0.0111	0.0124	0.0138	0.0154	0.0168	0.0201	0.0224	0.0248	0.0283	0.0303	0.0341	0.0050
4 to 8	4½, 5, 5½, 6	0.0063	0.0050	0.0067	0.0111	0.0124	0.0138	0.0154	0.0168	0.0201	0.0224	0.0248	0.0283	0.0303	0.0341	0.0050
8 to 10	7, 8, 9, 10	0.0063	0.0050	0.0067	0.0111	0.0124	0.0138	0.0154	0.0168	0.0201	0.0224	0.0248	0.0283	0.0303	0.0341	0.0050
10 to 16	11, 12, 14, 16	0.0063	0.0050	0.0067	0.0111	0.0124	0.0138	0.0154	0.0168	0.0201	0.0224	0.0248	0.0283	0.0303	0.0341	0.0050
16 to 24	18, 20, 22, 24	0.0063	0.0050	0.0067	0.0111	0.0124	0.0138	0.0154	0.0168	0.0201	0.0224	0.0248	0.0283	0.0303	0.0341	0.0050

for measurement of thread angles and pitch they should be held to close limits; see tables XIV.2, XIV.3, and XIV.4.

(c) *Tolerances on minor diameter of external thread and major diameter of internal thread.*—It will be sufficient in most instances to state only the maximum minor diameter of the external thread and the minimum major diameter of the internal thread without any tolerance. However, the root truncation from a sharp V should not be greater than $0.0826p$ or less than $0.0413p$.

7. MINIMUM CLEARANCES FOR EASY ASSEMBLY.—An allowance (clearance) should be provided on all buttress external threads in order to secure easy assembly of parts. The amount of the allowance should be deducted from the nominal major, pitch, and minor diameters of the external member in order to determine the maximum metal condition.

The minimum internal thread diameters will be basic.

The recommended allowance is the same for all three classes of thread and is equal to the class 3 (close) pitch diameter tolerance as calculated under par. 6(a), p. 29. The allowances for various combinations of pitch and diameter are given in table XIV.5.

The disposition of allowances and tolerances is indicated in figure XIV.2.

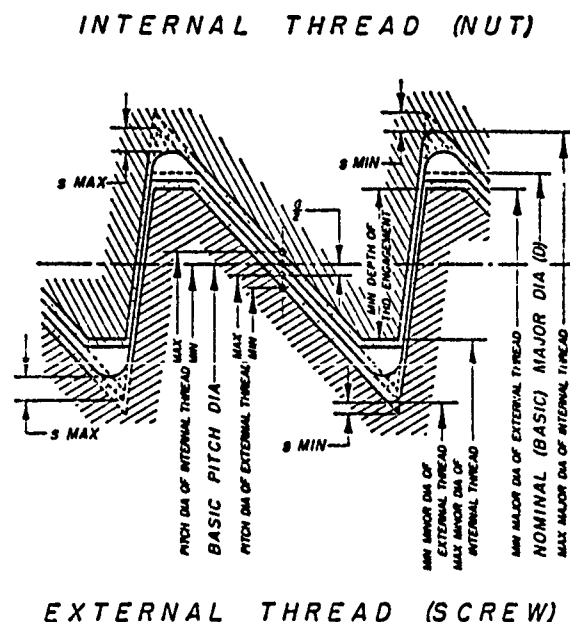


FIGURE XIV.2.—Illustration of tolerances, allowances, and root truncations, *Buttress threads*.

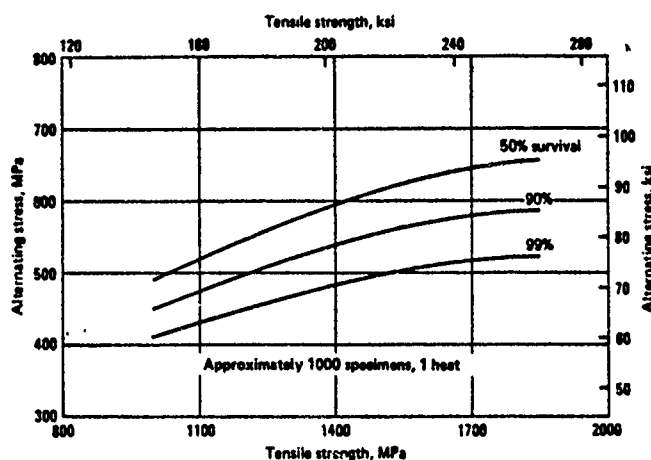
$\frac{G}{2} = \frac{1}{2}$ pitch diameter allowance on external thread
 s = root truncation

TABLE XIV.5.—Allowances on external Buttress threads, all classes

Major diameter	Preferred diameters	Threads per inch													
		20	16	12	10	8	6	5	4	3	2½	2	1½	1¼	1
		Allowance on major, minor, and pitch diameters													
in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.
½ to 1½	½, ¾, 1, 1½	0.0037	0.0040	0.0044	0.0049	0.0053	0.0058	0.0061	0.0065	0.0074	0.0077	0.0089	0.0100	0.0108	0.0135
1½ to 1	¾, ¾, 1	0.0042	0.0046	0.0051	0.0055	0.0060	0.0064	0.0068	0.0074	0.0077	0.0089	0.0100	0.0108	0.0135	0.0152
1 to 1½	1½, 1½, 1½, 1½	0.0043	0.0048	0.0051	0.0055	0.0060	0.0064	0.0068	0.0074	0.0077	0.0089	0.0100	0.0108	0.0135	0.0152
1½ to 2½	1½, 2, 2½, 2½	0.0046	0.0050	0.0053	0.0056	0.0061	0.0067	0.0071	0.0077	0.0089	0.0100	0.0108	0.0135	0.0152	0.0180
2½ to 4	2½, 3, 3½, 4	0.0049	0.0053	0.0056	0.0061	0.0067	0.0071	0.0077	0.0089	0.0100	0.0108	0.0135	0.0152	0.0180	0.0210
4 to 6	4½, 5, 5½, 6	0.0059	0.0064	0.0068	0.0072	0.0077	0.0083	0.0088	0.0094	0.0103	0.0109	0.0118	0.0130	0.0139	0.0152
6 to 10	7, 8, 9, 10	0.0063	0.0067	0.0070	0.0073	0.0078	0.0083	0.0088	0.0094	0.0103	0.0109	0.0118	0.0130	0.0139	0.0152
10 to 16	11, 12, 14, 16	0.0068	0.0072	0.0075	0.0078	0.0083	0.0088	0.0094	0.0103	0.0109	0.0118	0.0130	0.0139	0.0152	0.0180
16 to 24	18, 20, 22, 24	0.0077	0.0083	0.0088	0.0094	0.0103	0.0109	0.0118	0.0130	0.0139	0.0152	0.0180	0.0210	0.0240	0.0270

Screw-Thread Standards for Federal Services,
U. S. Department of Commerce. National
Bureau of Standards, 1966, Handbook H28
(1957) — Part III

Fig. 24 Scatter of fatigue limit data



Survival after 10 million cycles of AISI-SAE 4340 steel with tensile strengths of 995, 1320, and 1840 MPa (144, 191, and 267 ksi). Rotating-beam fatigue specimens tested at 10 000 to 11 000 rpm. Coefficients of variation range from 0.17 to 0.20.

value of b may be -0.1 . If the steel has been severely cold worked, the value of b may be -0.05 .

For a fatigue life of more than a million cycles, the use of these parameters in Eq 2 provides a slightly lower estimate of fatigue limit than the frequently used rule of thumb that the fatigue limit is half of the ultimate tensile strength.

The fatigue ductility coefficient, ϵ'_f , is approximated by the true fracture ductility, ϵ_f , which can be calculated from the reduction in area in a tension test by

$$\epsilon'_f \approx \epsilon_f = \ln \left(\frac{100}{100 - \%RA} \right) \quad (\text{Eq 6})$$

Typical values of ϵ'_f can be approximated from the Brinell hardness number as follows: ϵ'_f is 1.0 for HB less than 200; ϵ'_f is 0.5 for HB between 200 and 400; ϵ'_f is 0.1 for HB greater than 400. ϵ'_f should be calculated from %RA rather than using these approximate values, if possible.

The fatigue ductility exponent, c , has approximately the same value (-0.6) for most ductile steels. Severe cold working may reduce the value of c to -0.7 ; annealing or tempering at a high temperature may raise c to about -0.5 .

The elastic modulus (Young's modulus), E , is the slope of the elastic portion of the uniaxial stress/strain curve. For most steels, it has a value of about 200 GPa (29×10^6 psi). Further information on estimating these fa-

tigue parameters may be found in Ref 8.

Estimating Fatigue Life. Designers of machine components that will be subjected to cyclic loading would like to be able to predict the fatigue life from basic materials parameters and anticipated loading patterns. However, the scatter of fatigue data is so great that the likelihood of accurate predictions is extremely low. The methods and approximations in this article and Ref 4, 7, 8 and 12 can provide some indication of fatigue life.

In a particular situation, assessment of the seriousness of fatigue is aided by knowledge of the cyclic strains involved in fatigue at various lives. These generalizations are useful guidelines for ductile steels:

- 1 If the peak localized strains are completely reversed and the total range of strain is less than S_u/E , fatigue failures will occur in a large number of cycles or not at all.
- 2 If the total strain range is greater than 2% (amplitude $\pm 1\%$), fatigue failure will probably occur in less than 1000 cycles.
- 3 Part configurations that prevent utilization of the ductility of the metal or metals that have limited ductility are highly susceptible to fatigue failures.

In the long-life fatigue region, the relative magnitude of the change in fatigue strength due to processing may be crudely estimated by the relative

changes produced in the ultimate tensile strength and the hardness. If the ductility change is also measured and if the qualitative effects of various processes on different types of metal are known, more refined estimates of the change in fatigue behavior can be made without resorting to extensive fatigue testing.

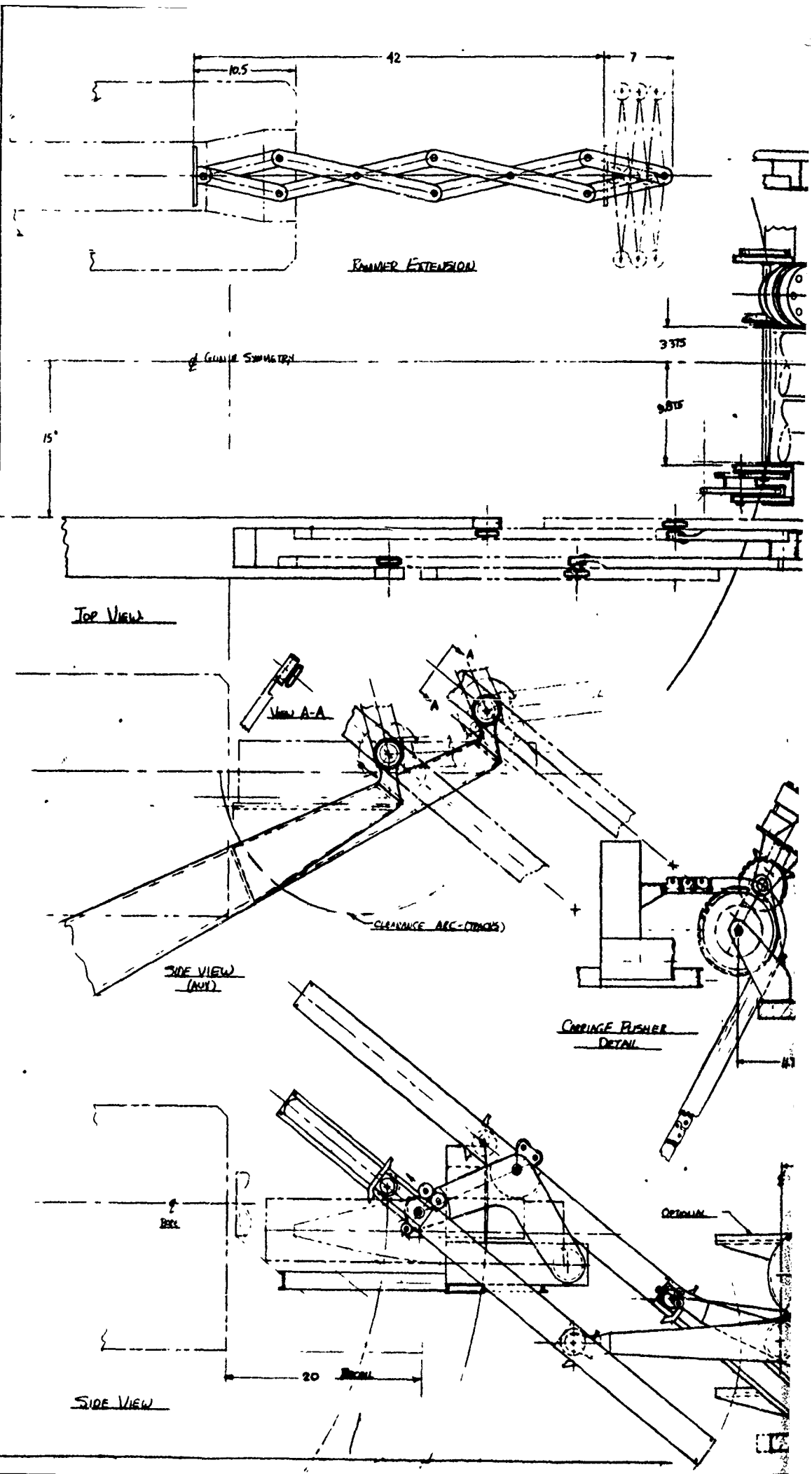
Fatigue life may be estimated by inserting a calculated strain amplitude and the appropriate materials parameters from Table 3 into Eq 4, solving for N_f . Where deformation is purely elastic, a calculated stress amplitude and Eq 2 may be used. The calculated fatigue life must be adjusted to compensate for stress concentrations, surface finish and the presence of aggressive environments, as described in Fig. 7 and Ref 2. Alternatively, the calculated stress may be adjusted by using stress concentration factors such as those in Ref 9 and 10. Any of these calculations include the assumption that the loading is fully reversed ($R = -1$).

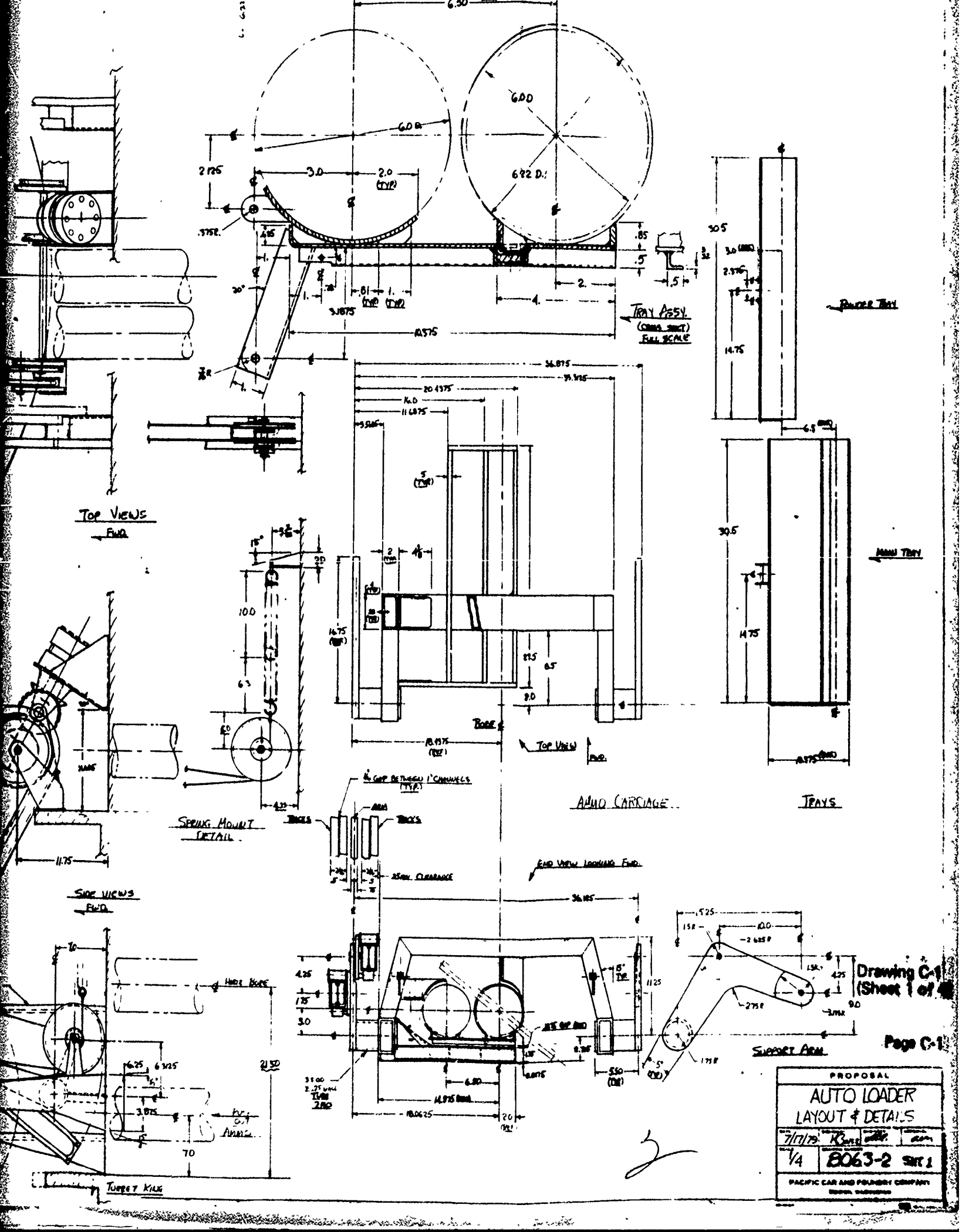
Potter (Ref 11) has described a method for approximating a constant-lifetime fatigue diagram for unnotched specimens. Using this method, a series of points corresponding to different lifetimes are calculated and plotted along the diagonal line on the left side ($R = -1$). Each of these points is connected by a straight line to the point of the other diagonal ($R = 1.0$) that corresponds to the ultimate tensile strength. A comparison between the estimated constant-lifetime diagram and the experimentally determined diagram is given in Fig. 26. The calculated lines correspond well with the experimental lines. Generally, the predicted lines represent lower stresses than the actual data. Estimating fatigue parameters from the Brinell hardness number provides more conservative estimates. These results are only approximations, and the methods may not apply for every material.

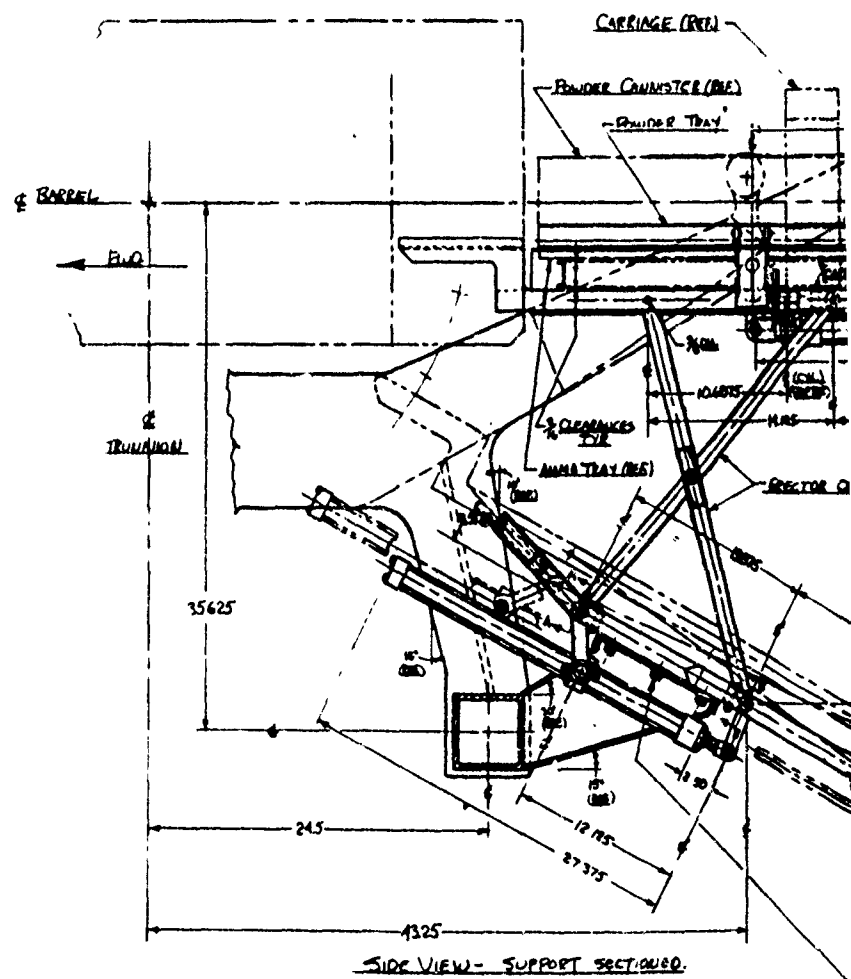
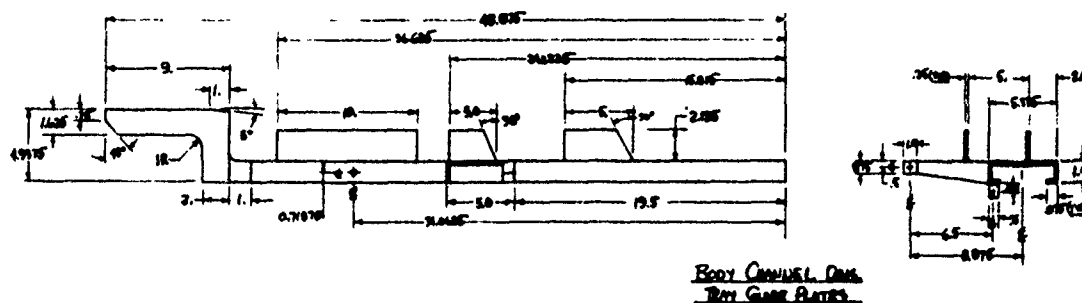
Cumulative Fatigue Damage.

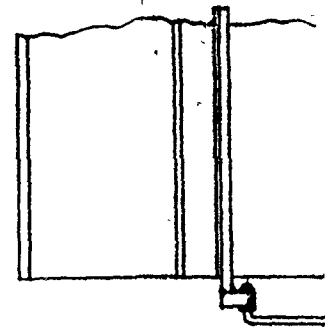
The data presented in this article, and most other published fatigue data, were obtained from constant-amplitude testing; every load cycle in the test is identical. In actual service, however, the loading can vary widely during the lifetime of a part. There have been many programs to evaluate the cumulative effects of variations in loading on the fatigue behavior of steels. References 3 and 11 through 13 describe methods of analyzing cumulative damage. A few overload cycles can reduce the fatigue

APPENDIX C
Engineering Drawings

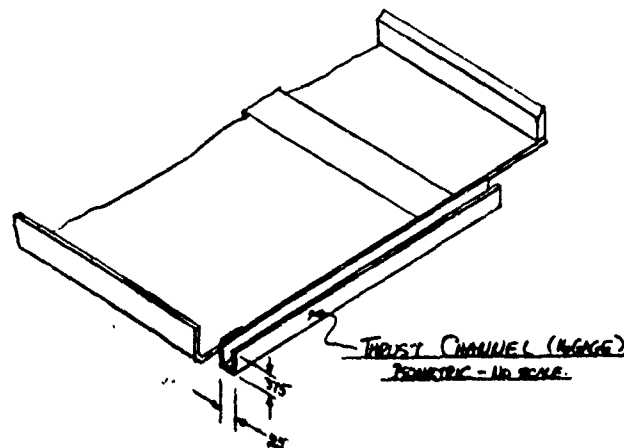






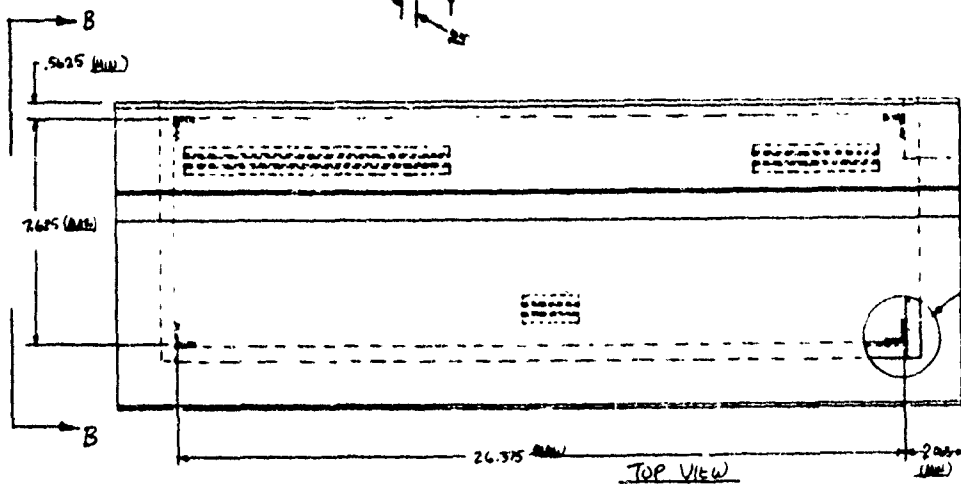


MAIN TRAY SHOWING SIDE RD



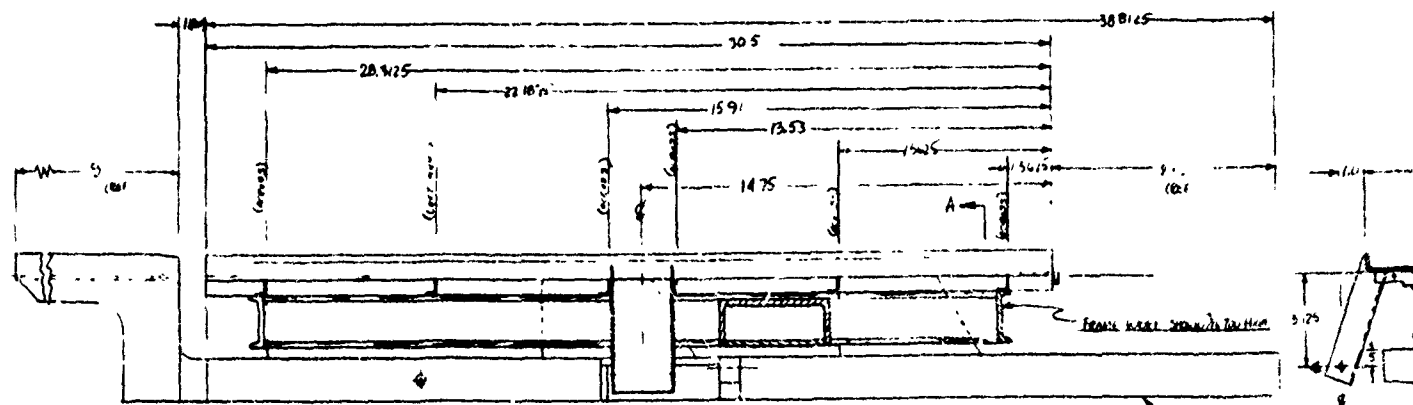
TRAY CHANNEL (HANGE)
ISOMETRIC - NO SCALE

VIEW B-B
ROTATED 90°
FULL SIZE

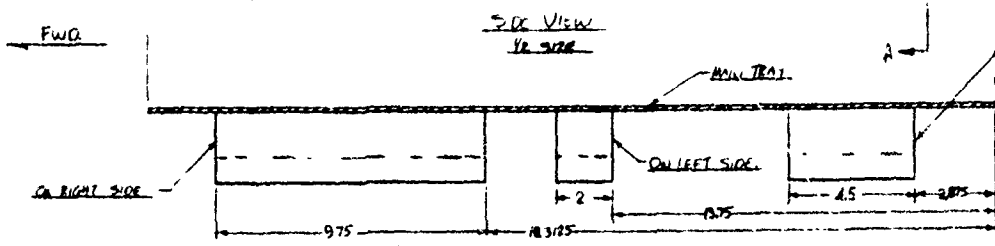


TOP VIEW

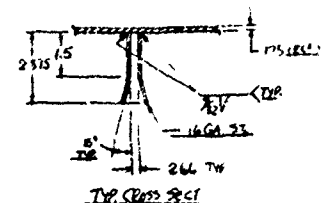
(1) POSITIONING ANGLE IN EACH OF (4) CORNERS



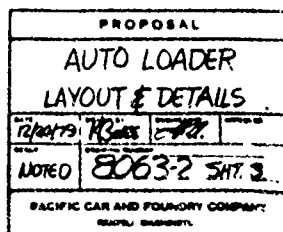
SIDE VIEW
1/2 SIZE



SECT SIDE VIEW
1/2 SIZE



TOP CROSS SECT



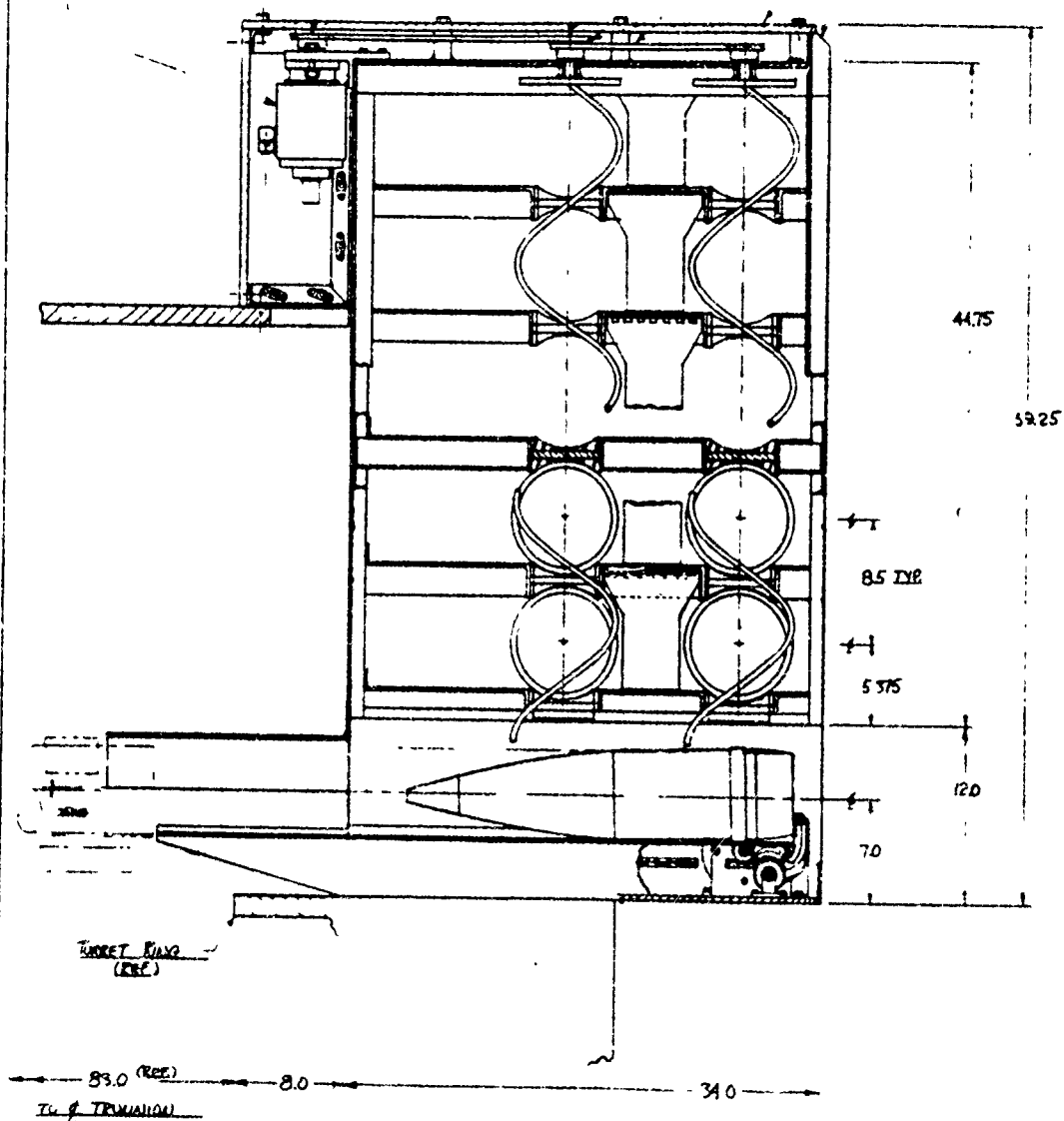
- SPROCKET LINK-BELT RC 35
TYPE B, PATCH, DOUBLE WIDTH
21 TEETH, PD 2.54 IN. OR EQUIV.

- PARKER ROTATING MOTOR
1500 PSI PRESSURE
SPEED 6-100 RPM
TORQUE 385 IN-LBS/100 PSI

- SPROCKET-LINK-BELT
RC 35 TYPE B, 3/4 DIA
21 TEETH, PD 2.54 IN.
OR EQUIV. (2) REQD.

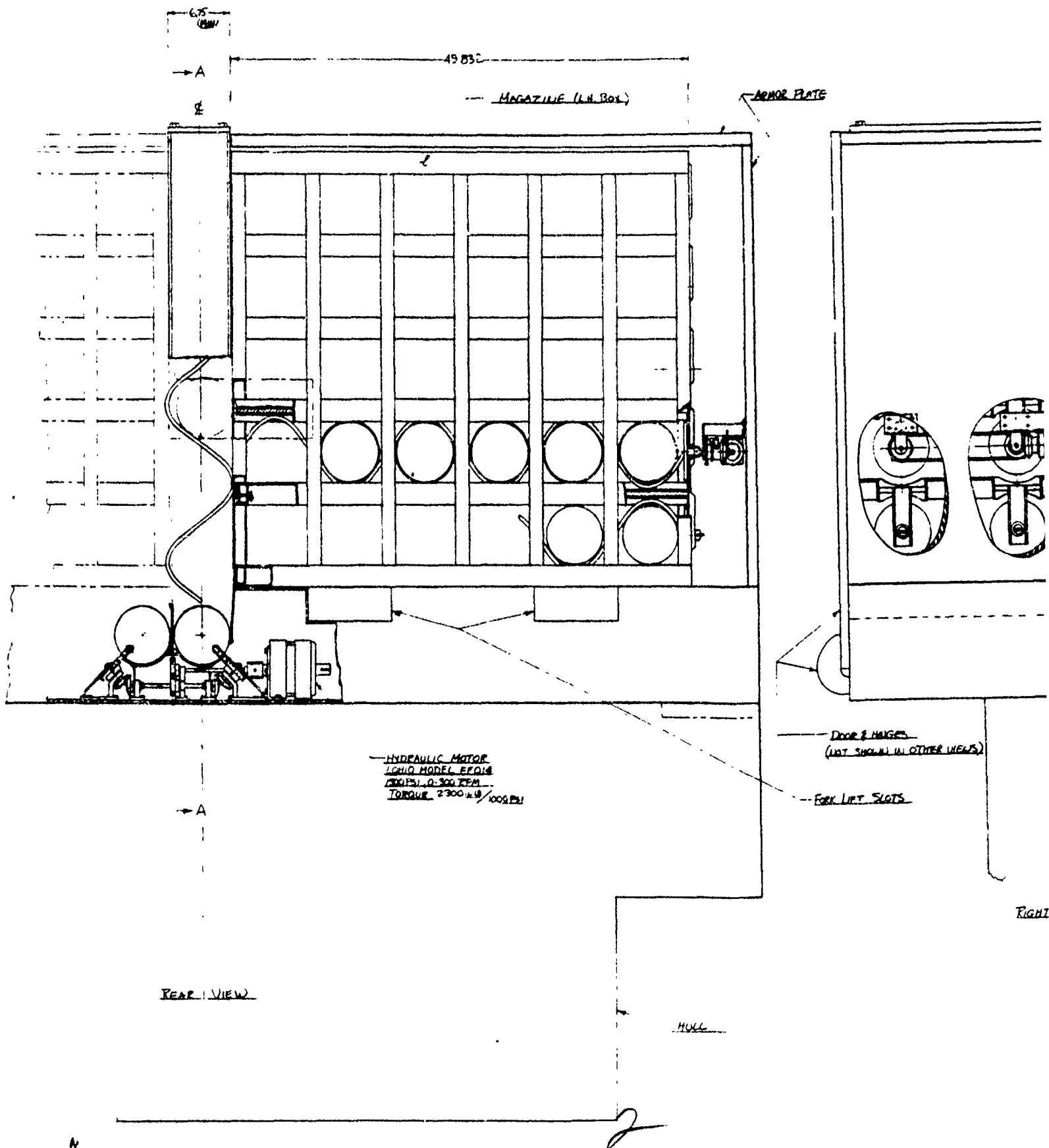
- CHAIN LINK-BELT RC 35
PATCH, 3/4 IN. DIA. 12000

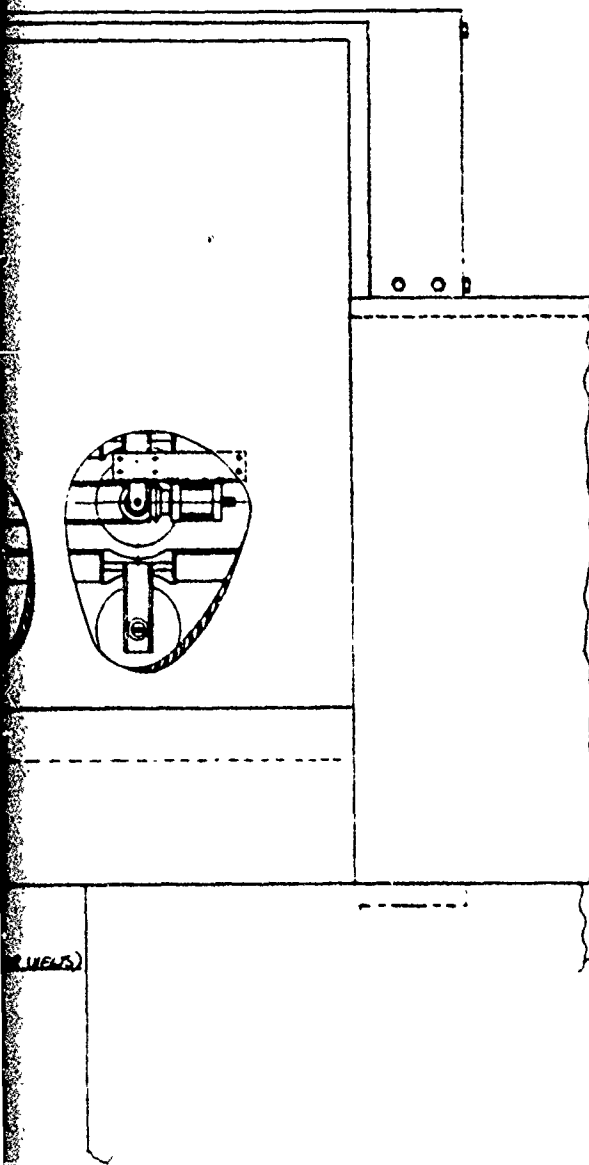
- ELEVATOR SECT.



SECTION A-A

FWO LEFT SIDE VIEW





RIGHT SIDE VIEW →

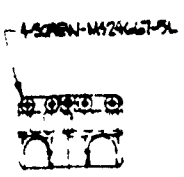
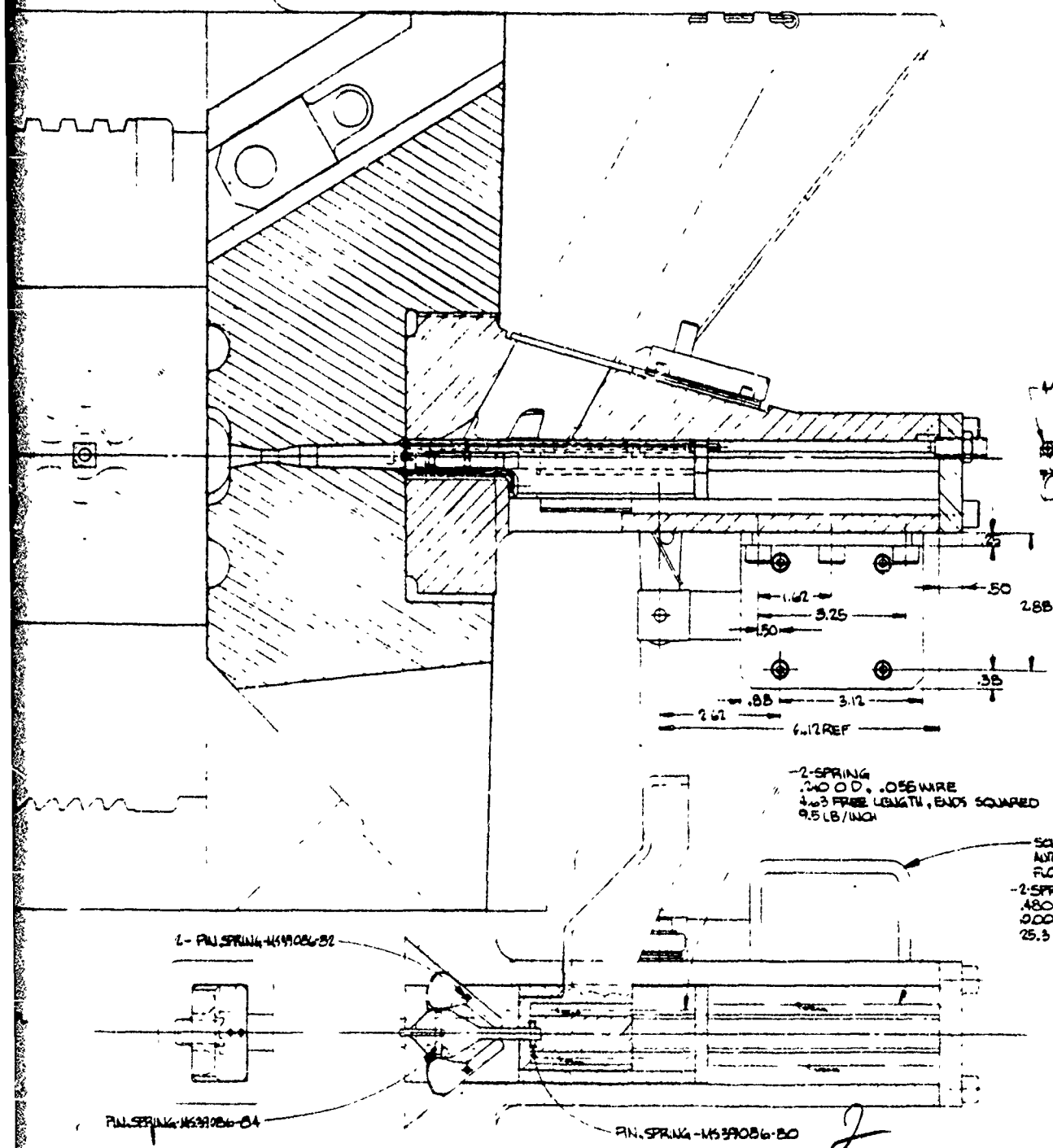
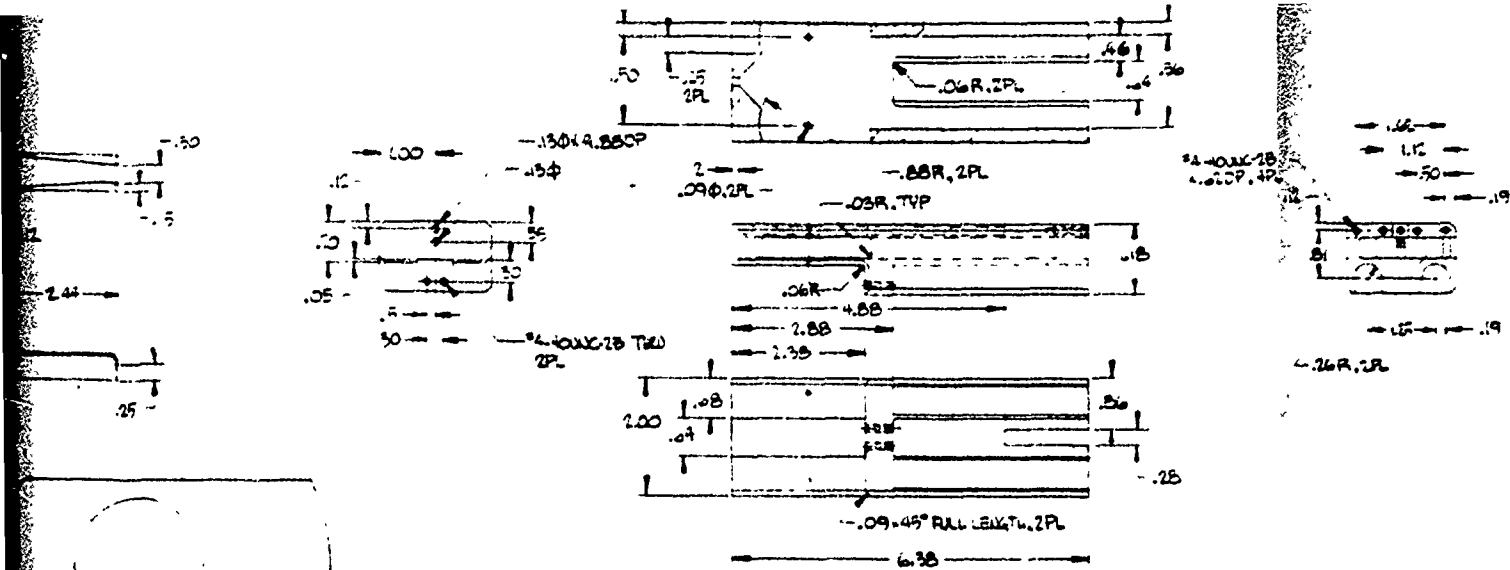
Drawing C-1
(Sheet 4 of 4)

Page C-4

PROPOSAL			
AUTO LOADER			
LAYOUT - MAGAZINE			
2-22-80	K. B. Smith	W. H. Smith	adm.
1/4	3063-2	SHT. 4	
PACIFIC CAR AND FOUNDRY COMPANY			
TACOMA, WASHINGTON			

3





2-SPRING
 .240 O.D., .055 WIRE
 4.63 FREE LENGTH, ENDS SQUARED
 9.5 LB/INCH

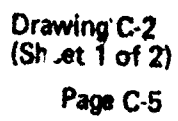
SOLENOID #10651060
 AUTOMATIC SWITCH CO.
 FLORHAM PARK, N.J. 07932
 2-SPRING
 .480 O.D., .067 WIRE
 10.00 FREE LENGTH, ENDS SQUARED
 25.3 LB/INCH

1- PIN SPRING - MS39086-02

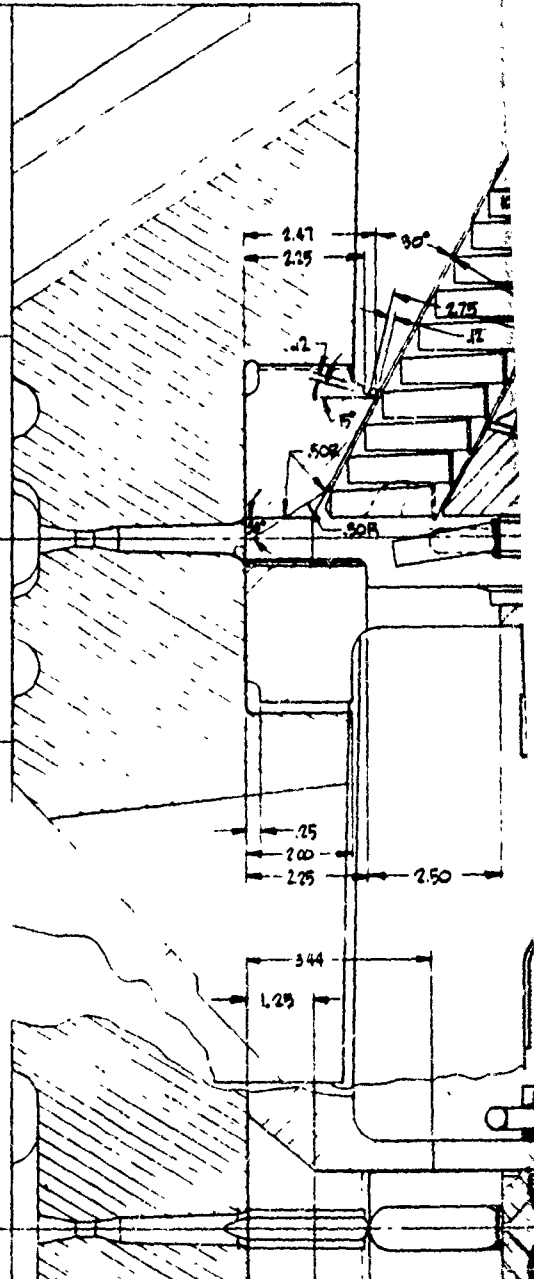
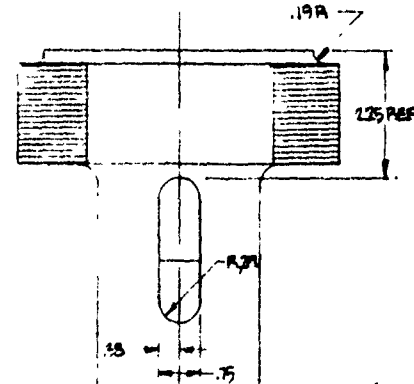
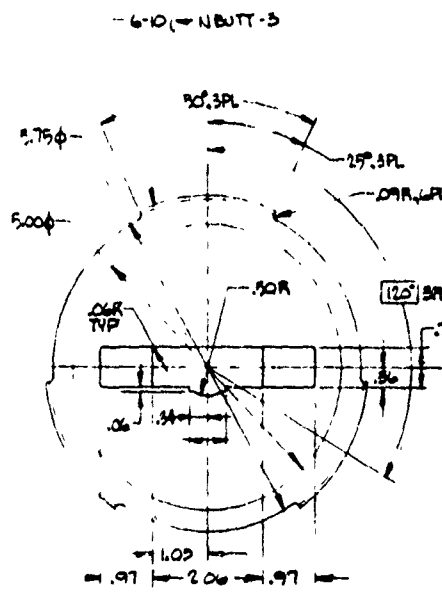
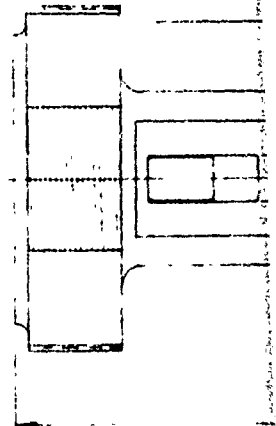
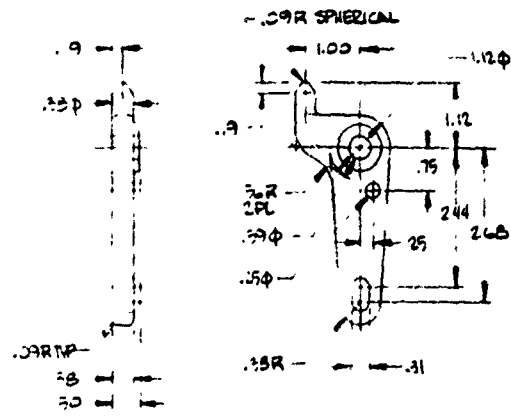
PIN SPRING - MS39086-04

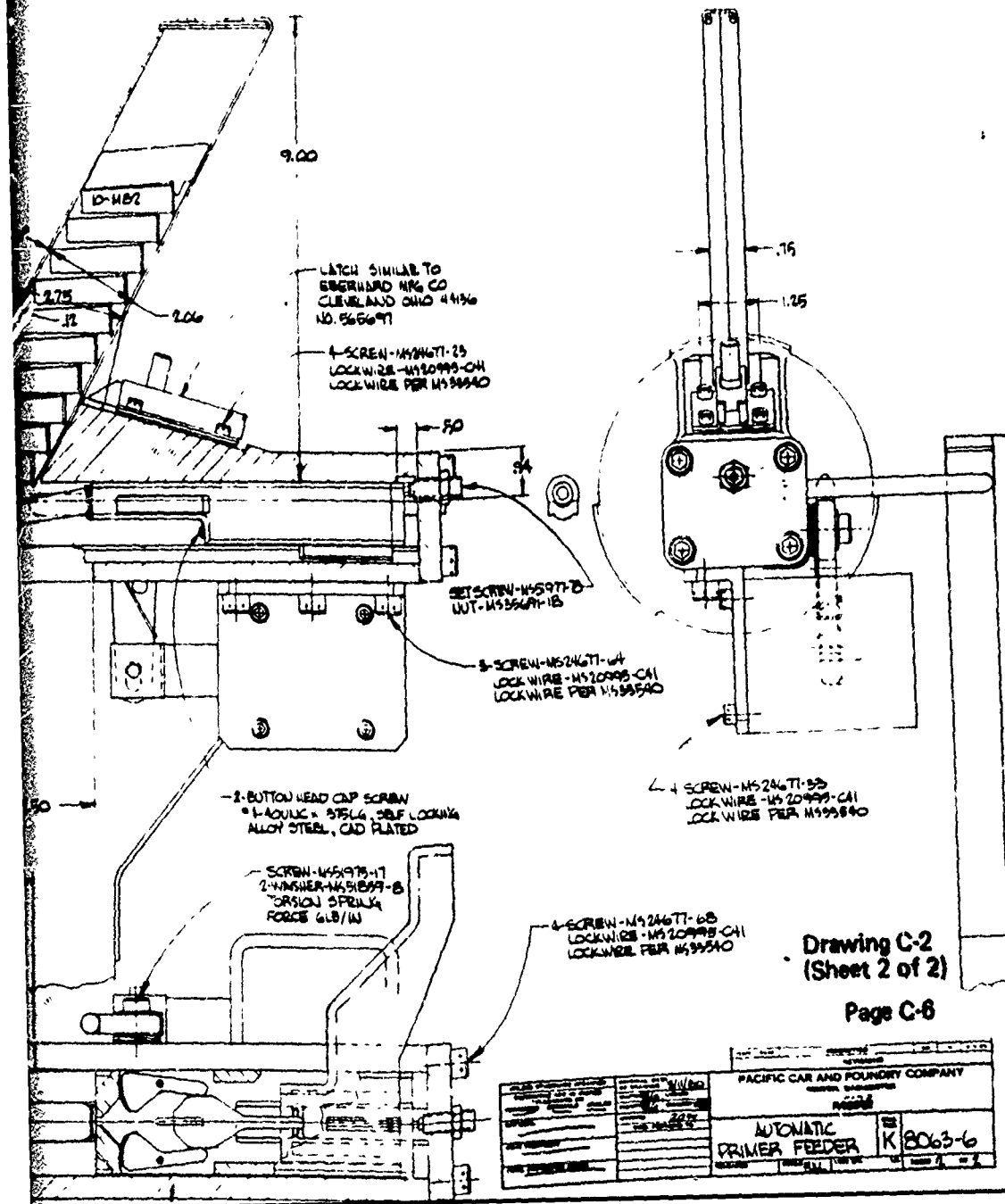
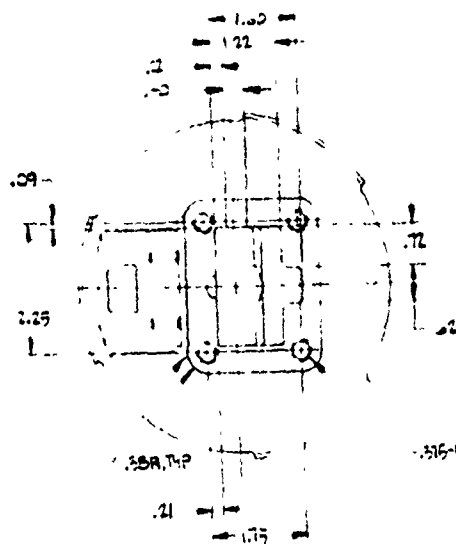
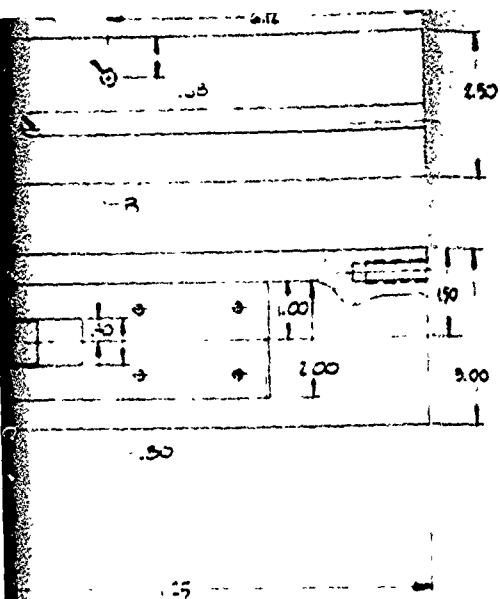
PIN SPRING - MS39086-00

2



1. NAME OF THE COMPANY 2. ADDRESS 3. CITY 4. STATE 5. ZIP 6. PHONE 7. FAX 8. E-MAIL 9. WEBSITE 10. OTHER	11. NAME OF THE PERSON 12. ADDRESS 13. CITY 14. STATE 15. ZIP 16. PHONE 17. FAX 18. E-MAIL 19. WEBSITE 20. OTHER	21. NAME OF THE COMPANY 22. ADDRESS 23. CITY 24. STATE 25. ZIP 26. PHONE 27. FAX 28. E-MAIL 29. WEBSITE 30. OTHER
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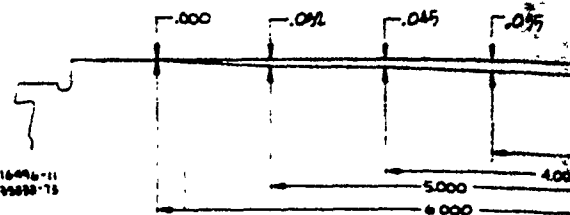
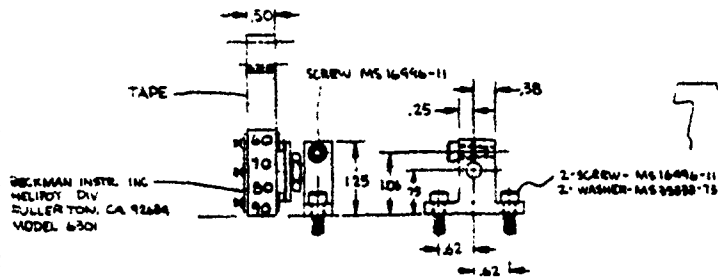
Drawing C-2
(Sheet 2 of 2)

Page C-6

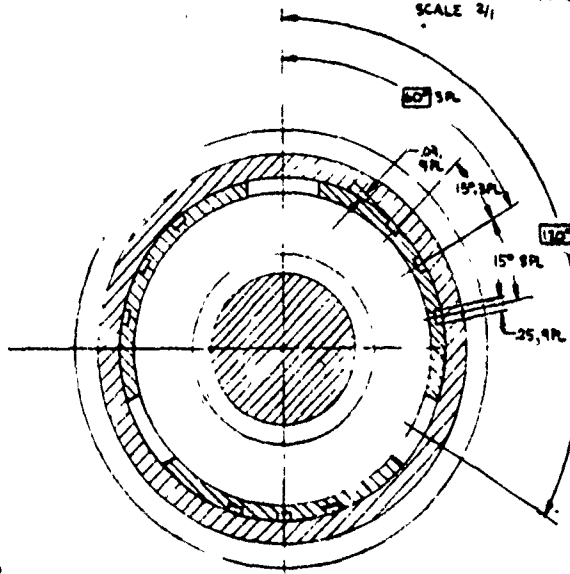
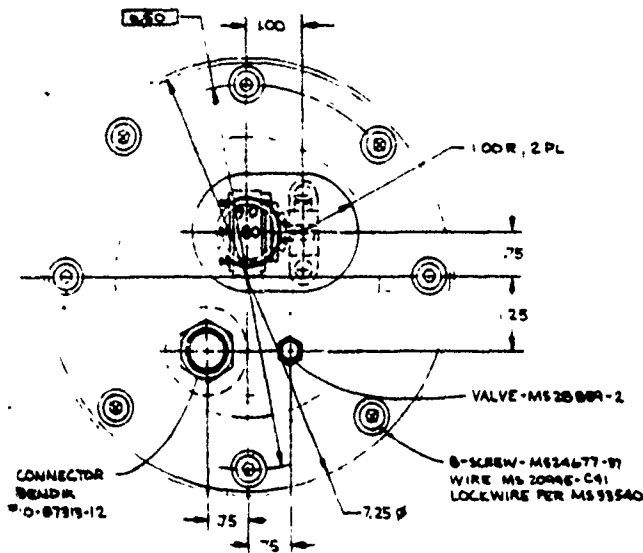
PACIFIC CAR AND FOUNDRY COMPANY	
AUTOMATIC PRIMER FEEDER	
K 8063-6	

2

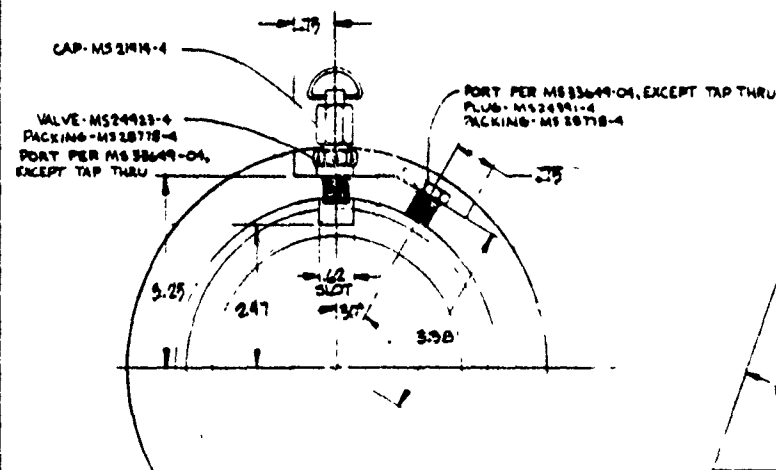
BEKMAN INSTR. INC.
WILLYT DIV.
FULLERTON, CA 92634
MODEL 6301



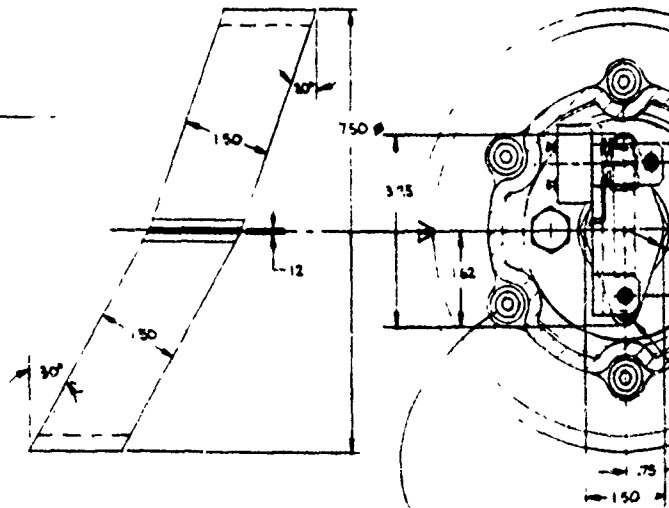
DETAIL BUFFER, S.P.
SCALE 2/1



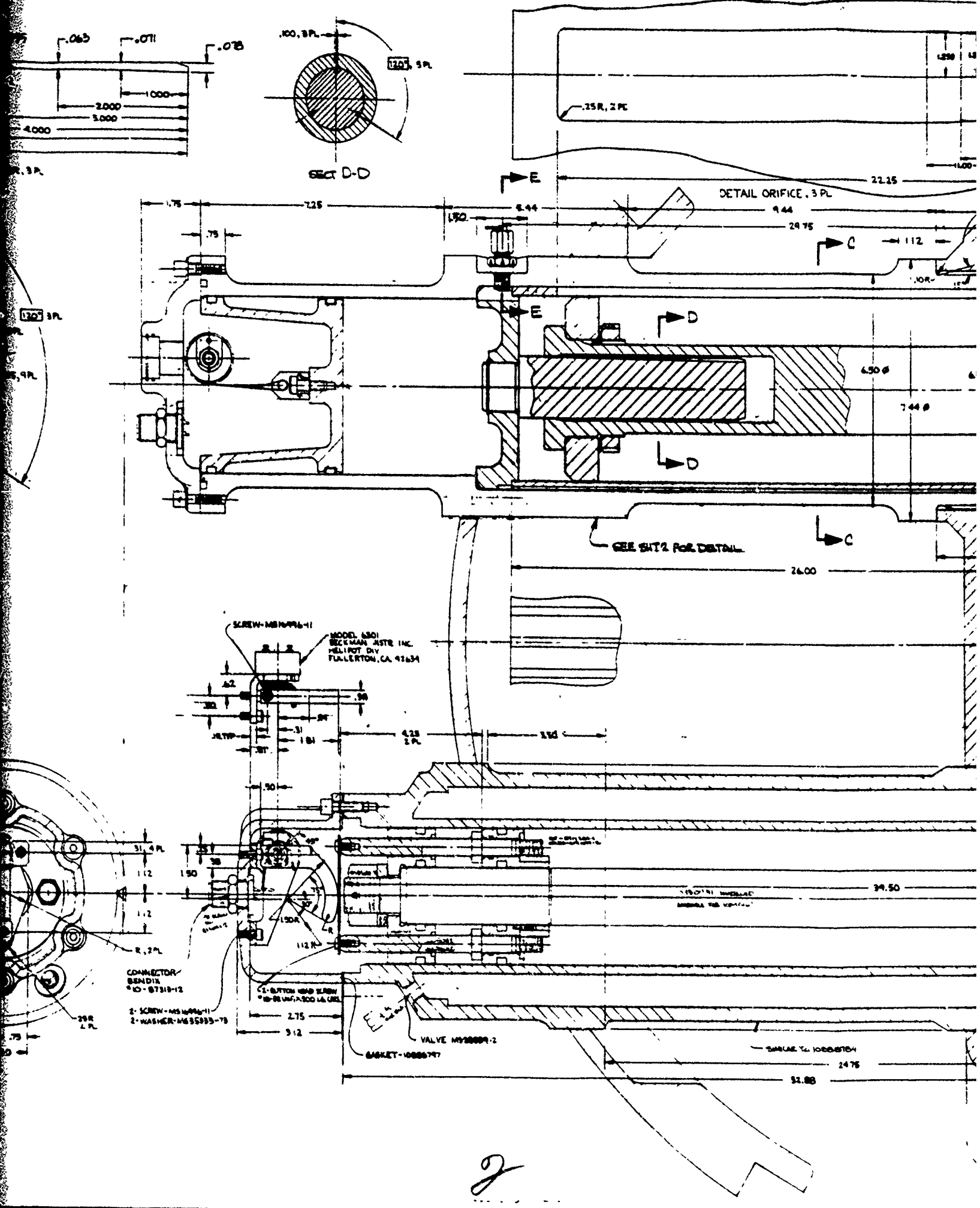
SECT C-C

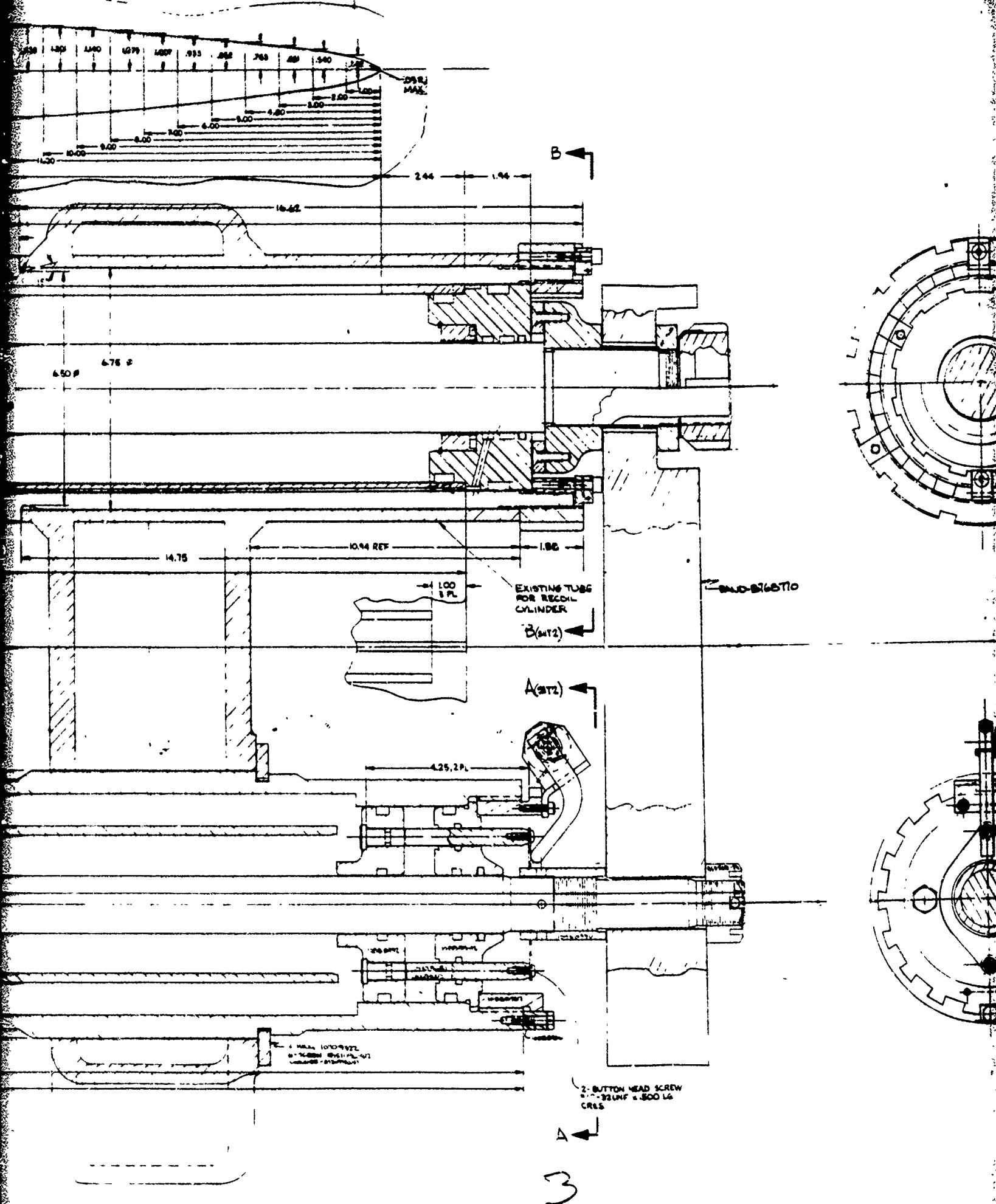


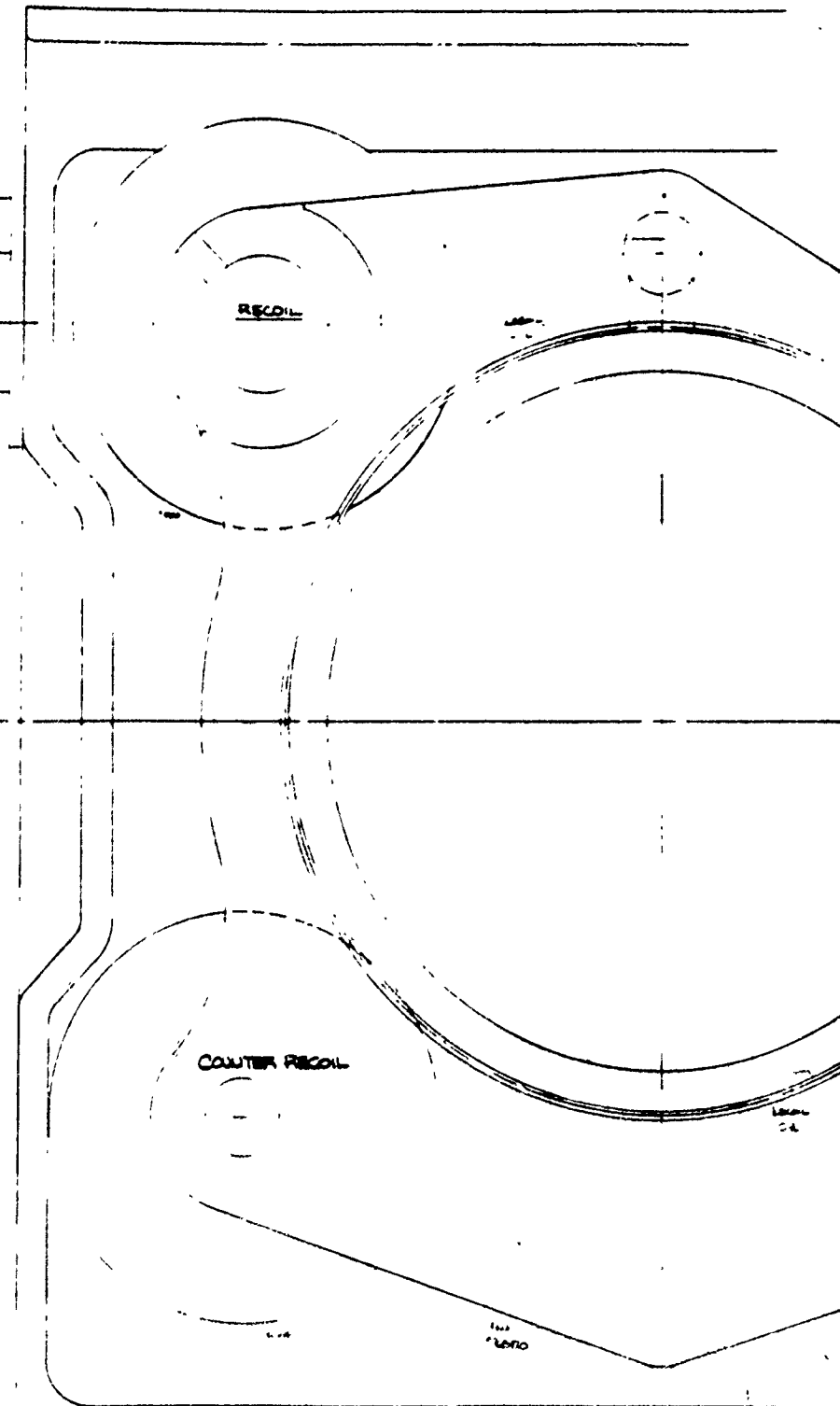
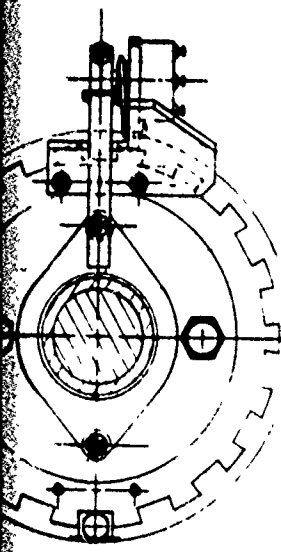
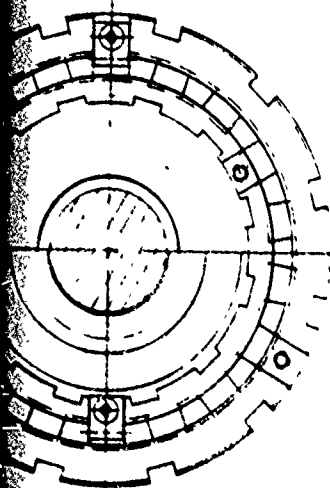
SECT E-E



8 SCREW - MS 24677-37
WIRE - MS 20986 C41
LOCKWIRE PER MS 93540







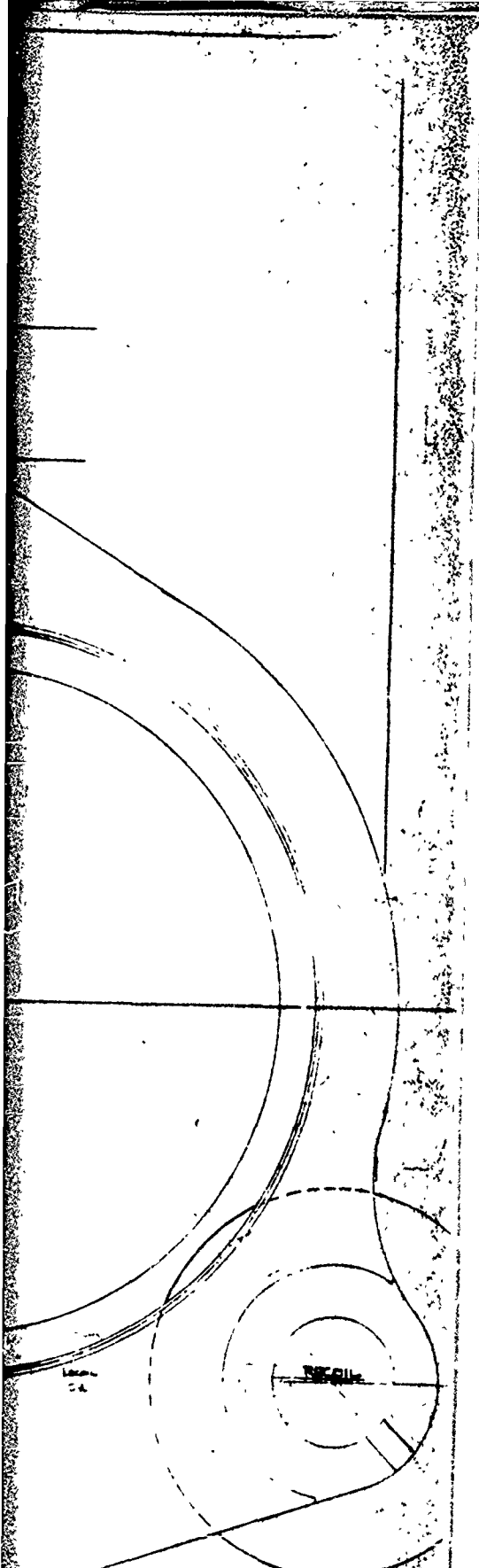
RECOIL

COUNTER RECOIL

RECOIL

4

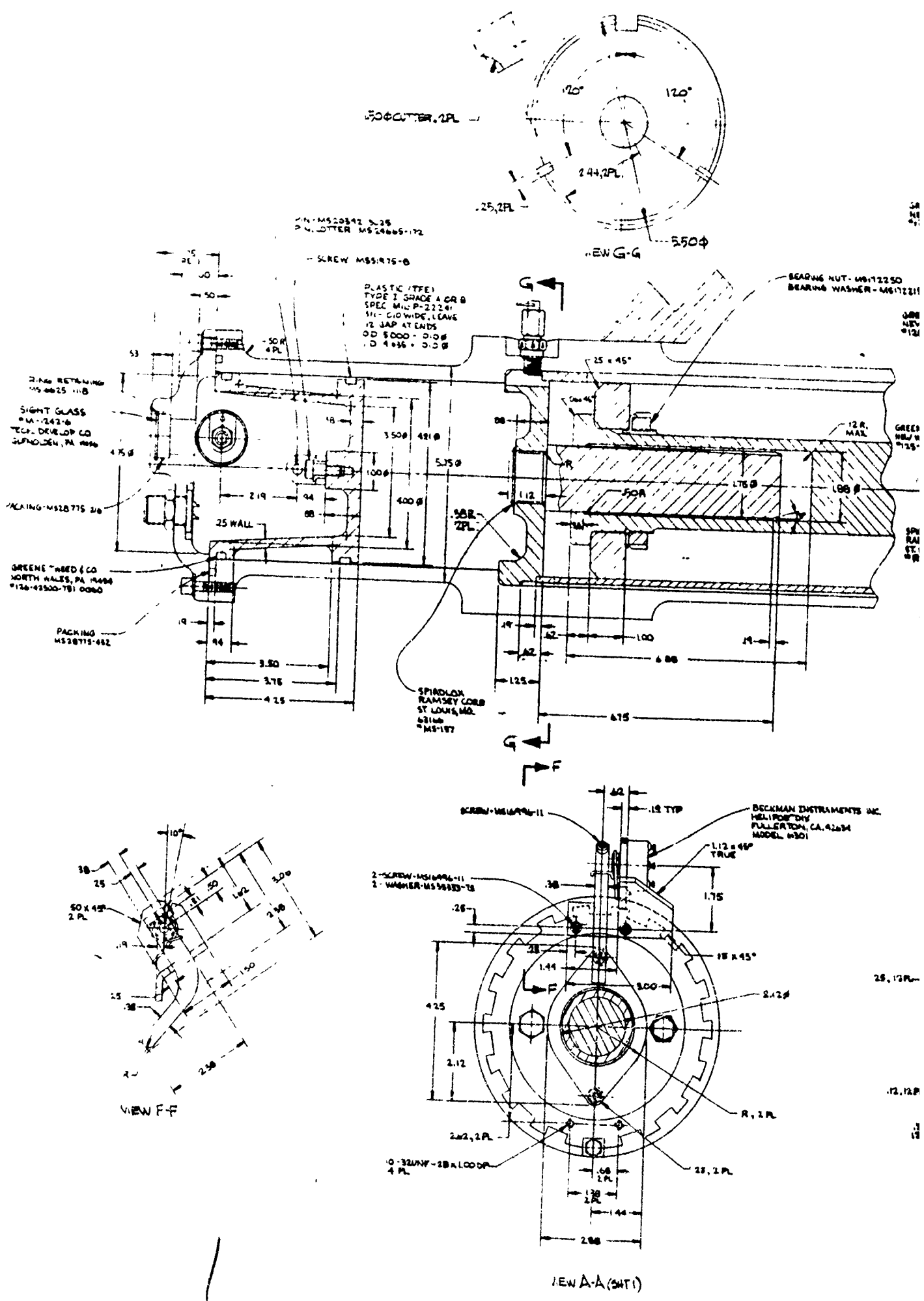
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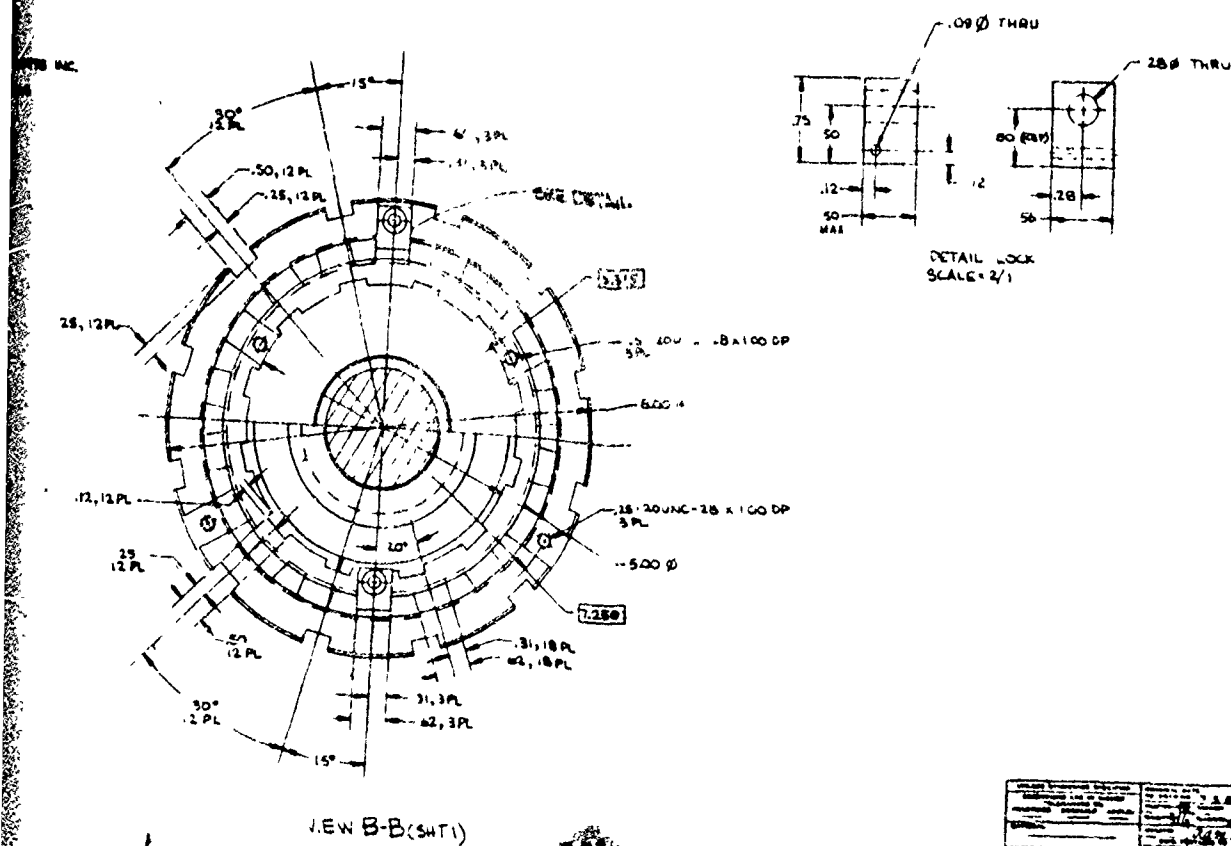


Drawing C-3
Sheet 1 of 2

Page C-7

PACIFIC CAR AND FOUNDRY COMPANY	
155 MM MODULAR RECOIL SYSTEM	
K	0063-5





Page C-8

NAME: <u>James Earl Ray</u> ADDRESS: <u>1000 1st St. S.W.</u> CITY: <u>Atlanta, Ga.</u> STATE: <u>Ga.</u> ZIP: <u>30303</u>		PHONE: <u>404-525-1234</u> FAX: <u>404-525-1234</u> E-MAIL: <u>James.E.Ray@att.net</u>	
COMPANY: <u>PACIFIC CAR AND POLYGRAPH COMPANY</u> ADDRESS: <u>1000 1st St. S.W.</u> CITY: <u>Atlanta, Ga.</u> STATE: <u>Ga.</u> ZIP: <u>30303</u>		PHONE: <u>404-525-1234</u> FAX: <u>404-525-1234</u> E-MAIL: <u>James.E.Ray@att.net</u>	
PRODUCT: <u>155 MM MODULAR RECOIL SYSTEM</u> ADDRESS: <u>1000 1st St. S.W.</u> CITY: <u>Atlanta, Ga.</u> STATE: <u>Ga.</u> ZIP: <u>30303</u>		PHONE: <u>404-525-1234</u> FAX: <u>404-525-1234</u> E-MAIL: <u>James.E.Ray@att.net</u>	
ORDER: <u>155 MM MODULAR RECOIL SYSTEM</u> ADDRESS: <u>1000 1st St. S.W.</u> CITY: <u>Atlanta, Ga.</u> STATE: <u>Ga.</u> ZIP: <u>30303</u>		PHONE: <u>404-525-1234</u> FAX: <u>404-525-1234</u> E-MAIL: <u>James.E.Ray@att.net</u>	

NOTES

1. SPRING DATA

DIAMETER OF WIRE

(A)

1.50

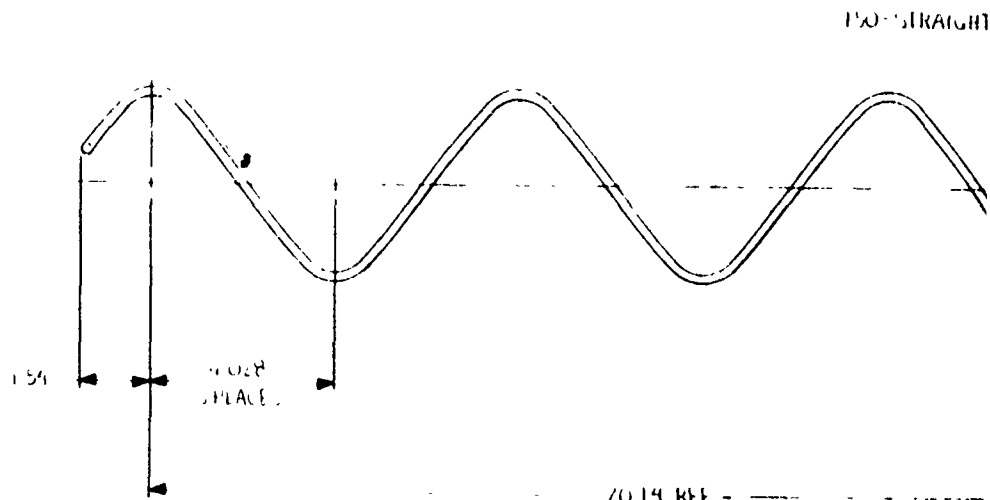
DIRECTION OF HELIX

RIGHT HAND

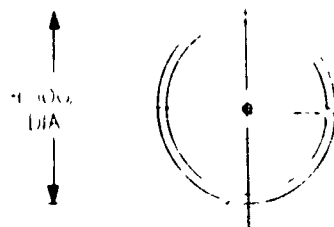
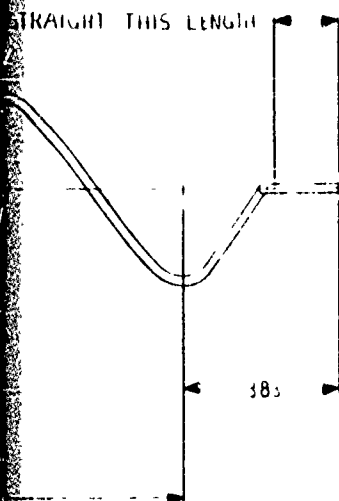
2. MATERIAL MUSIC STEEL, SPRING WIRE, FINISH
BRIGHT, ASTM A228.

3. FINISH PLATING, CADMIUM TYPE II, CLASS OPTIONAL,
SPEC QQ P 416.

4. TOLERANCE ON ALL TWO PLACE DECIMALS $\pm .005$
TOLERANCE ON ALL THREE PLACE DECIMALS $\pm .001$

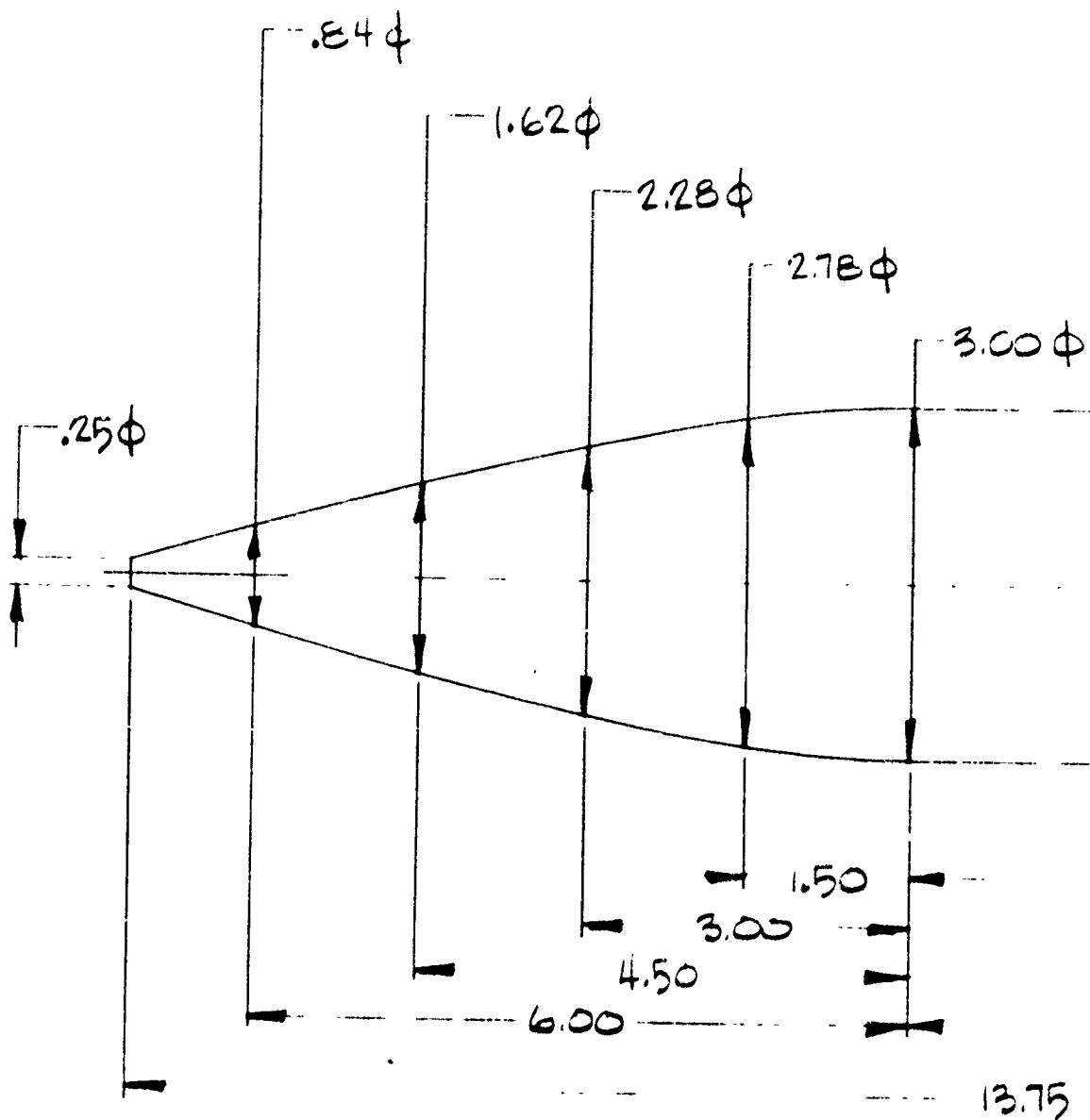


2014 REF



Drawing C-1
Page C-1

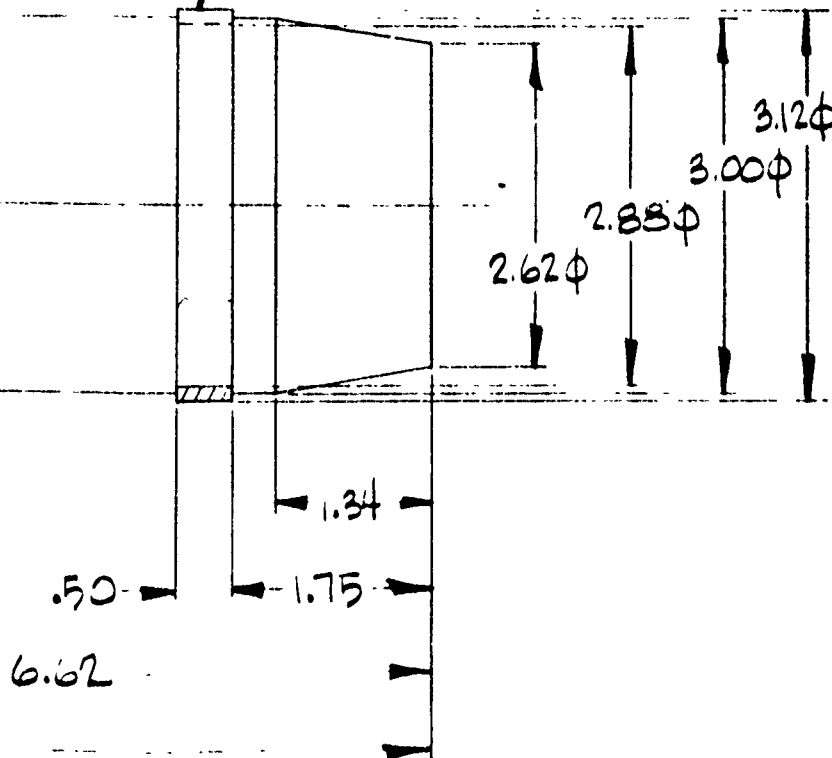
		A 9.117 WA'S 177 ISSUE DATE DESCRIPTION	101 102
ORIGINAL DATE OF DRAWING 8-14-79 DRAWN BY <i>HL</i> CHECKED DIMENSIONS ARE IN INCHES FRACTIONS DECIMALS ANGLES TOLERANCES ON SEE NOTE 4		REVISIONS PACIFIC CAR AND FOUNDRY COMPANY BENTON, WASHINGTON A DIVISION OF INCO	
MATERIAL SEE NOTE 1 HEAT TREATMENT SEE NOTE 3 FINAL PROTECTIVE FINISH SEE NOTE 3		APPROVED BY <i>HL</i> DWS. PERTAINS TO AUTOLOADER PICK UP	DWS. NO. D FEEDER, SCREW 8063-
RELEASED		SCALE 1/2	UNIT WT -- LB. QTY 1



UNLESS OTHERWISE SPECIFIED:		ORIGINAL
DIMENSIONS ARE IN INCHES		OF DRAWING
TOLERANCES ON		DRAWN BY
FRACTIONS	DECIMALS	SUBMITTER
$\pm .00$		
MATERIAL		APPROVED
STEEL		DWG. NO.
HEAT TREATMENT		155111
FINAL PROTECTIVE FINISH		
PAINT O.D.		

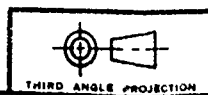
BILL OF MATERIAL			
ITEM NO.	QTY	NAME	SPECIFICATION

COPPER NUTS;
JOINT LOCATION; OPTIONAL
OPTIONAL MAT'L:
ALUMINUM



Drawing C-5

Page C-10



ISSUE	DATE	DESCRIPTION	DR.	CHK.	E. N. NO.
-------	------	-------------	-----	------	-----------

REVISIONS

PACIFIC CAR AND FOUNDRY COMPANY
RENTON, WASHINGTON
A DIVISION OF
PACOR

DESIGNED	ORIGINAL DATE OF DRAWING	11/1/79
DRAWN	CHECKED	
SUBMITTED	EXAMINED	
APPROVED		
DWG. PERTAINS TO		
155MM AUTOMATED		

155MM PROJECTILE
(STEEL MOCK-UP)

DWG. SIZE

C

2063-4

RELEASED

SCALE FULL

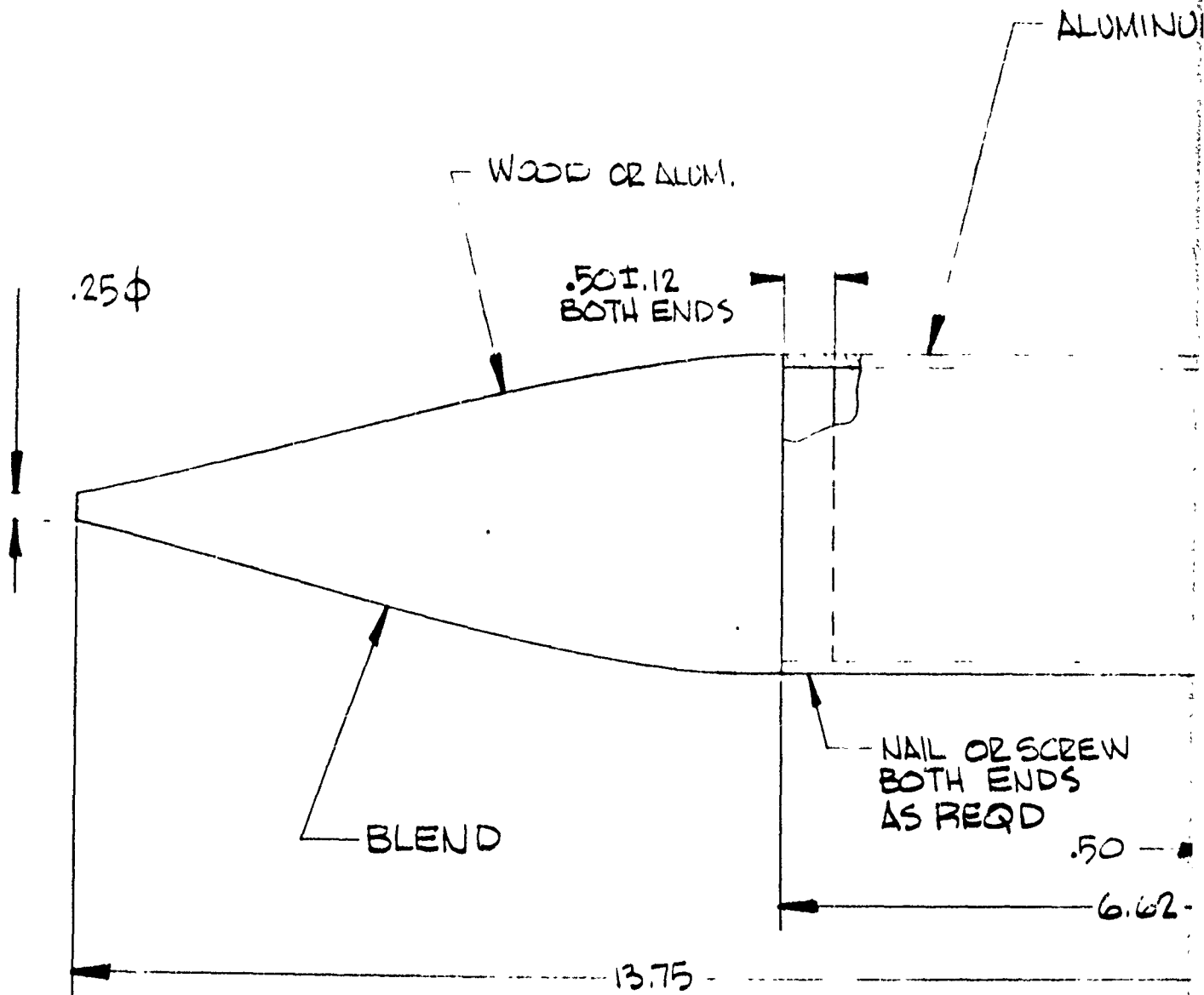
UNIT WT

18.5 LB.

SHEET

OF

2

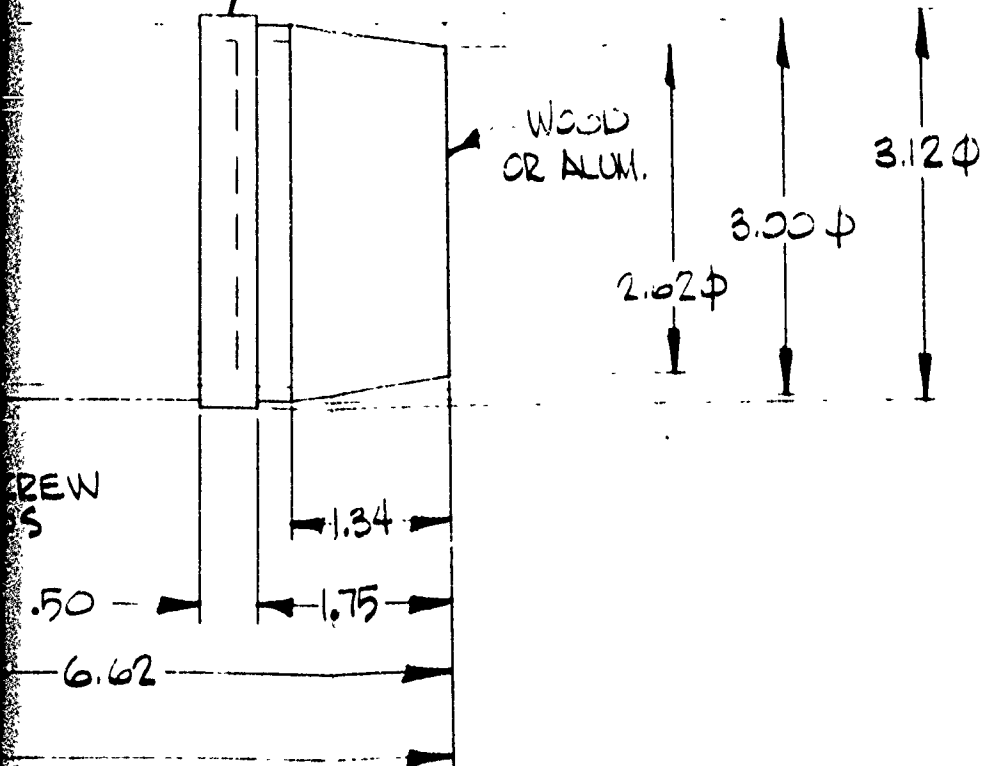


UNLESS OTHERWISE SPECIFIED			ORH
DIMENSIONS ARE IN INCHES			OF
TOLERANCES ON			DRY
FRACTIONS	DECIMALS	ANGLES	SUBS
	$\pm .06$		
MATERIAL			APP
WOOD & ALUM			15
HEAT TREATMENT			
FINAL PROTECTIVE FINISH			
PAINT O.D.			

BILL OF MATERIAL			
ITEM NO.	QTY	NAME	SPECIFICATION

ALUMINUM .125 WALL

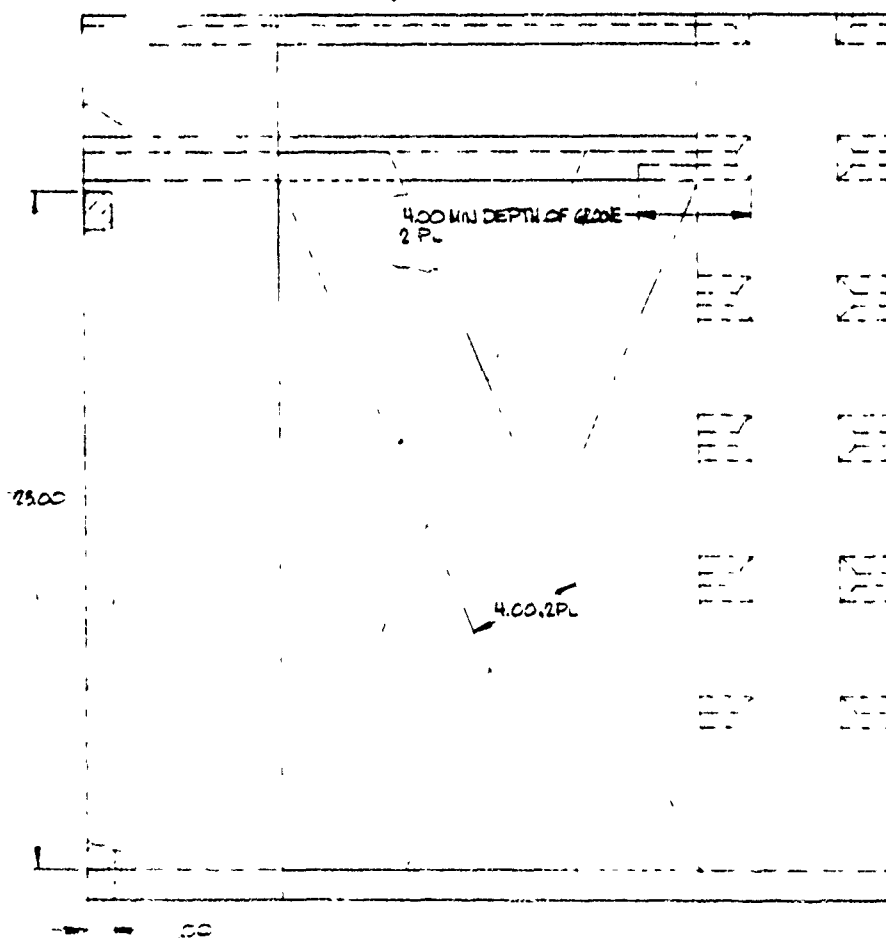
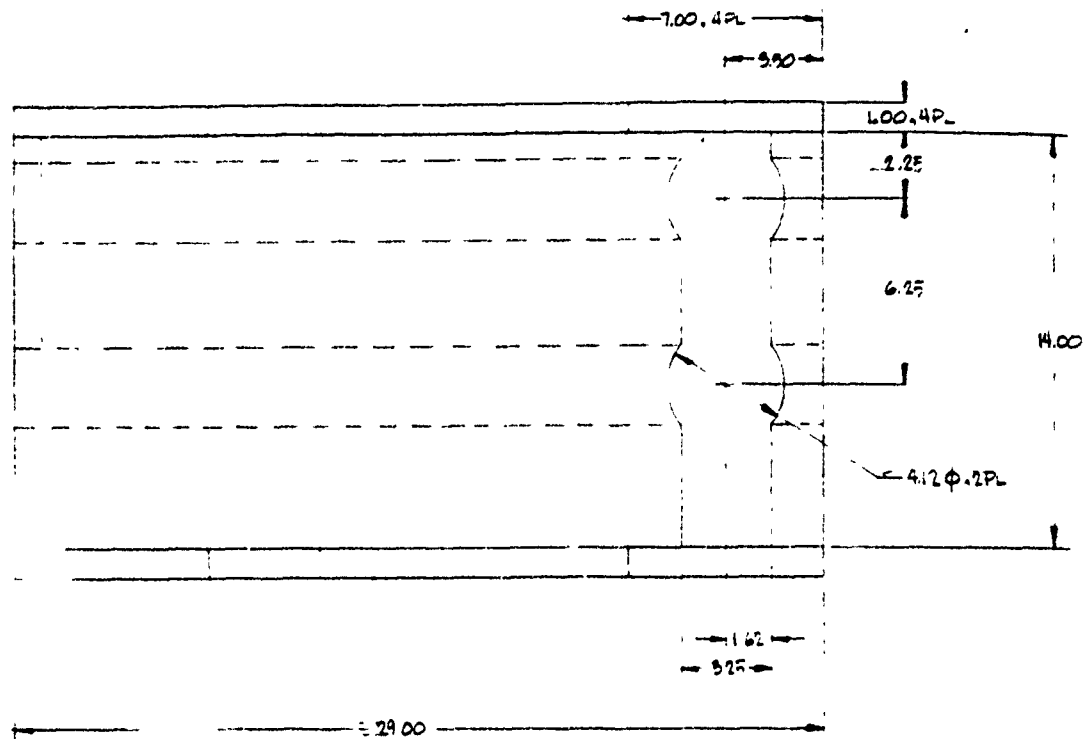
OPTIONAL:
COPPER OR ALUMINUM BAND

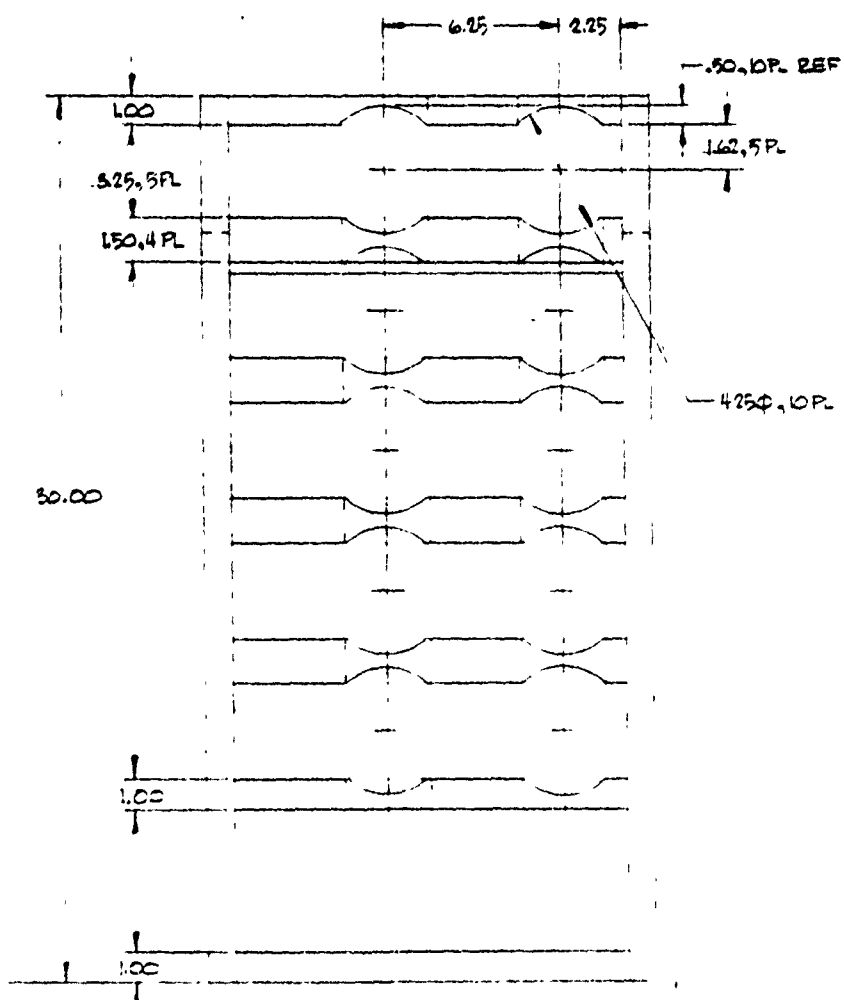


Drawing C-6.

Page C-11

 THIRD ANGLE PROJECTION		ISSUE DATE DESCRIPTION DR. CH. S. N. NO.
REVISIONS		
PACIFIC CAR AND FOUNDRY COMPANY RENTON, WASHINGTON A DIVISION OF PACOR		
SPECIFIED IN INCHES ON ANGLES	ORIGINAL DATE OF DRAWING 12/2/71 DRAWN BY G. CHECKED BY SUBMITTED BY EXAMINED BY APPROVED BY	DWG. PERTAINS TO 155 MM SHELL (MOCK-UP)
ALUM	DWG. SIZE C 8063-8	RELEASED SCALE FULL UNIT WT L.B. SHEET OF



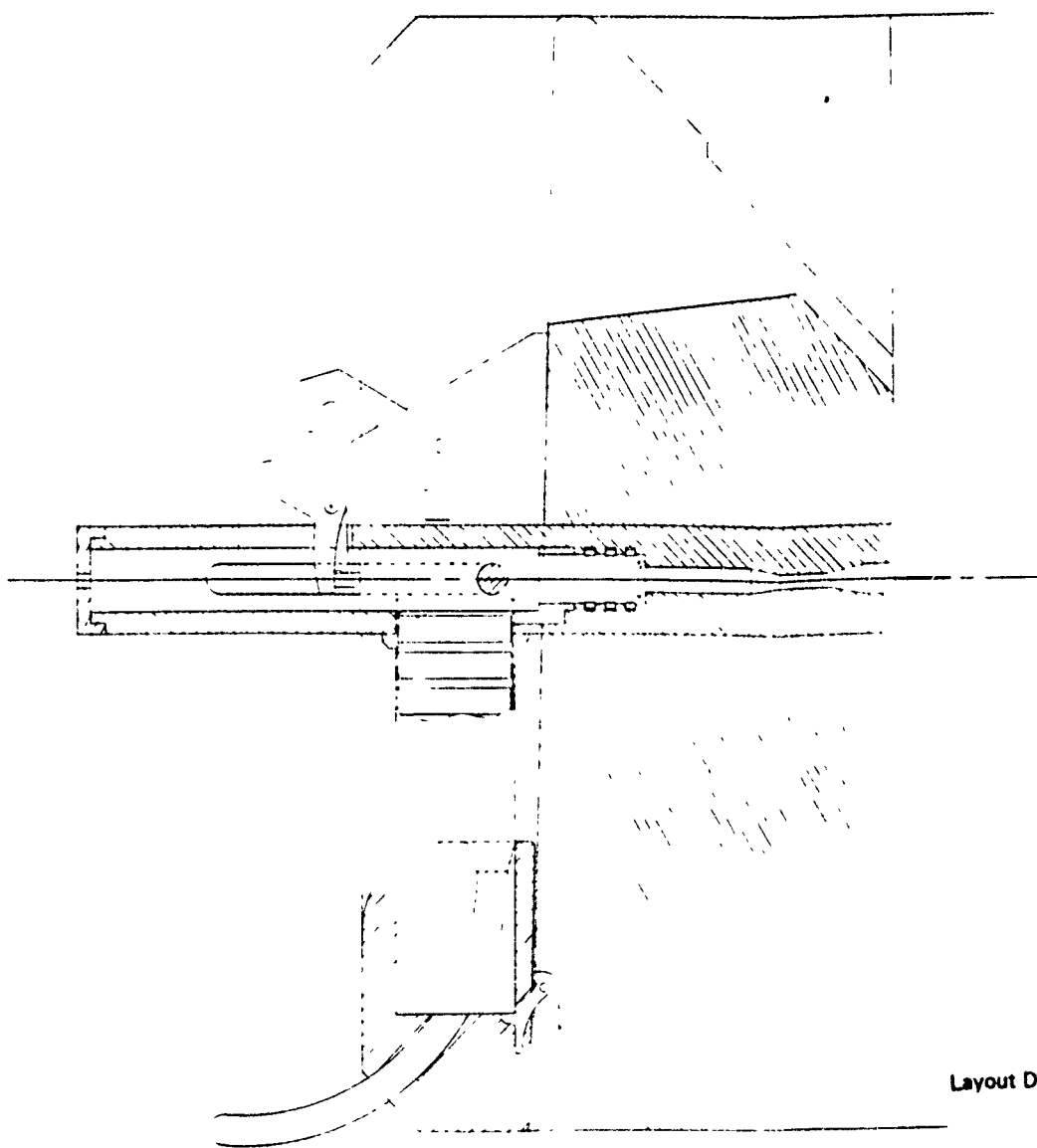


6. TO ASSEMBLE USE GLUE, NAILS AND/OR SCREWS
AS REQUIRED

Page C-12

[illegible]

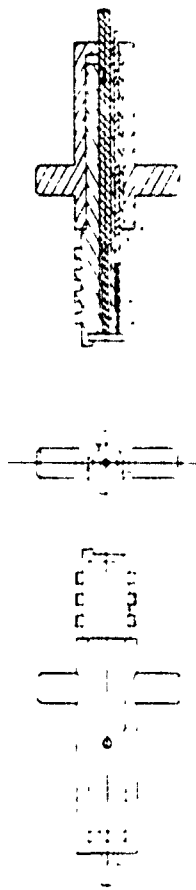
APPENDIX D
Engineering Layouts



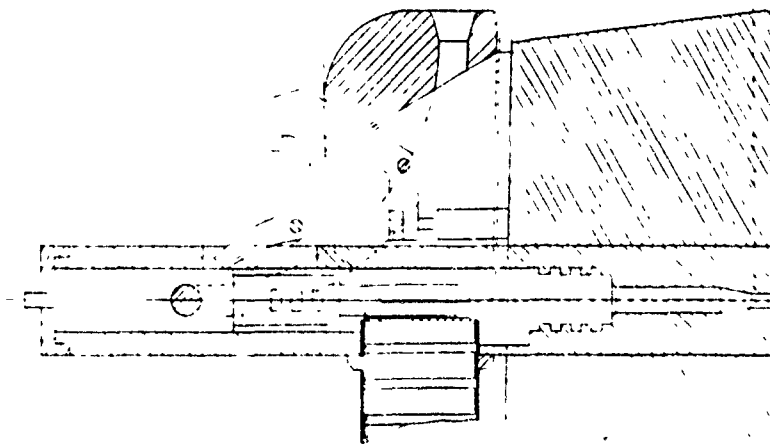
Layout D-1. Rotary Bolt Primer Feeder — Breech Closed

SCALE 1/1

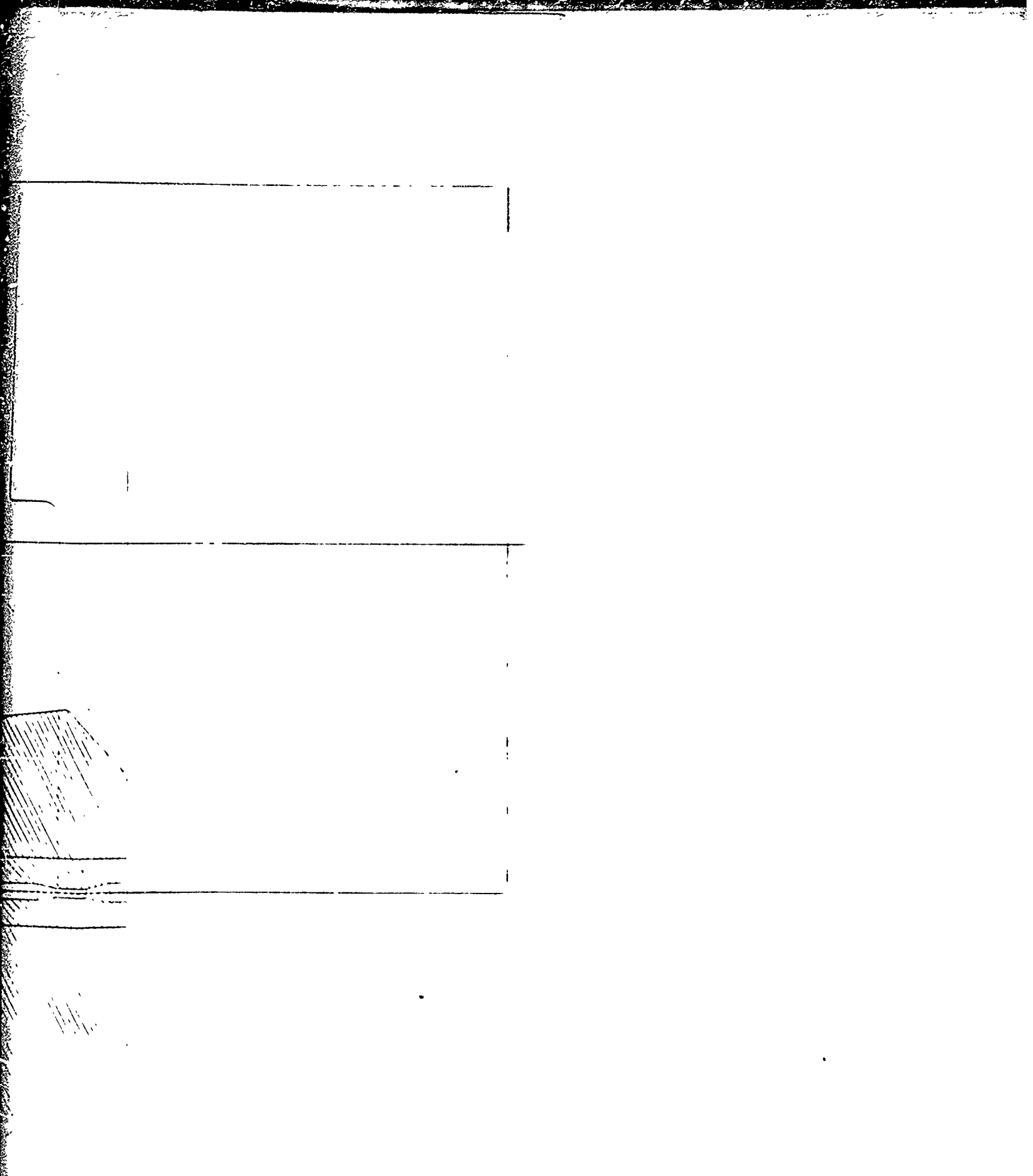
SECTION THROUGH CENTERLINE OF BREECH



BOLT AND BOLT CARRIER



SECTION THROUGH CENTERLINE OF BARREL

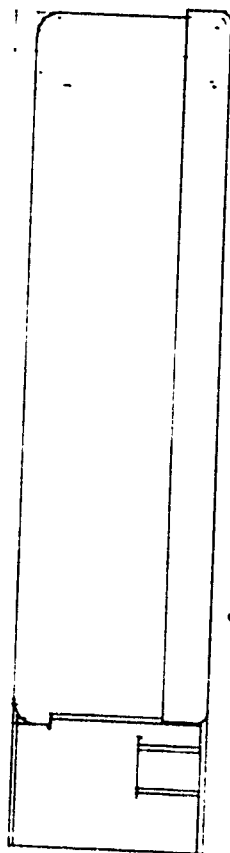


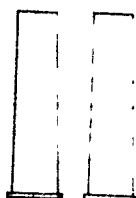
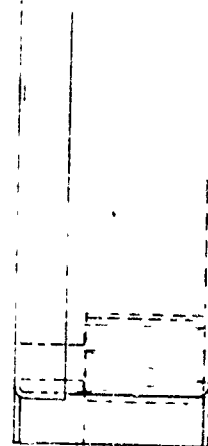
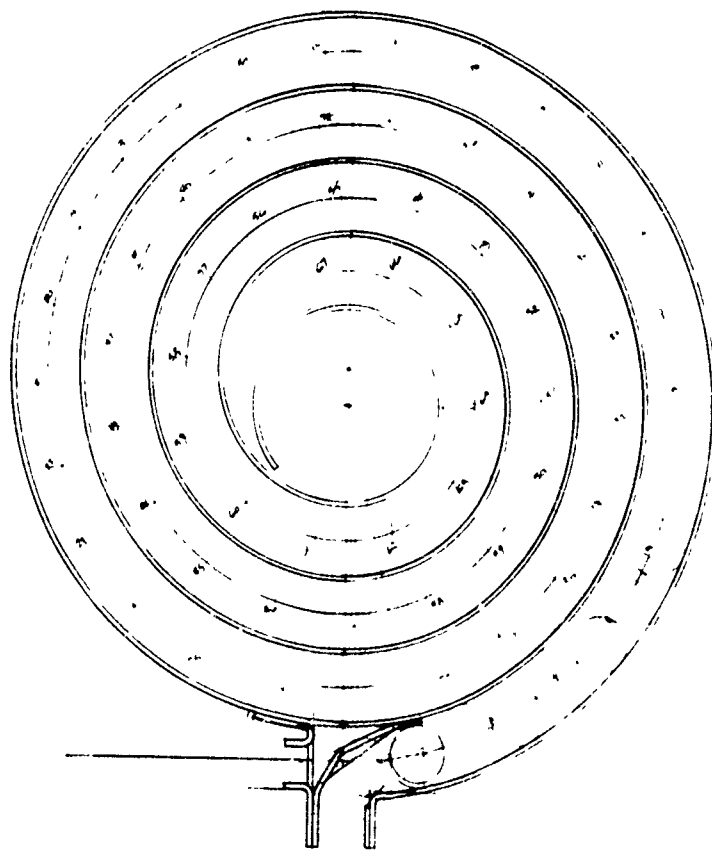
Layout D-2. Rotary Bolt Primer Feeder -- Breech Open

SCALE 1/1

2







Layout D-3. 60-Round Primer Drum

2